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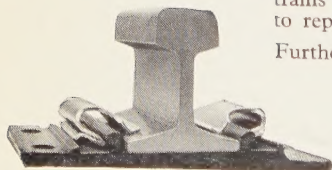
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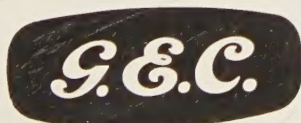
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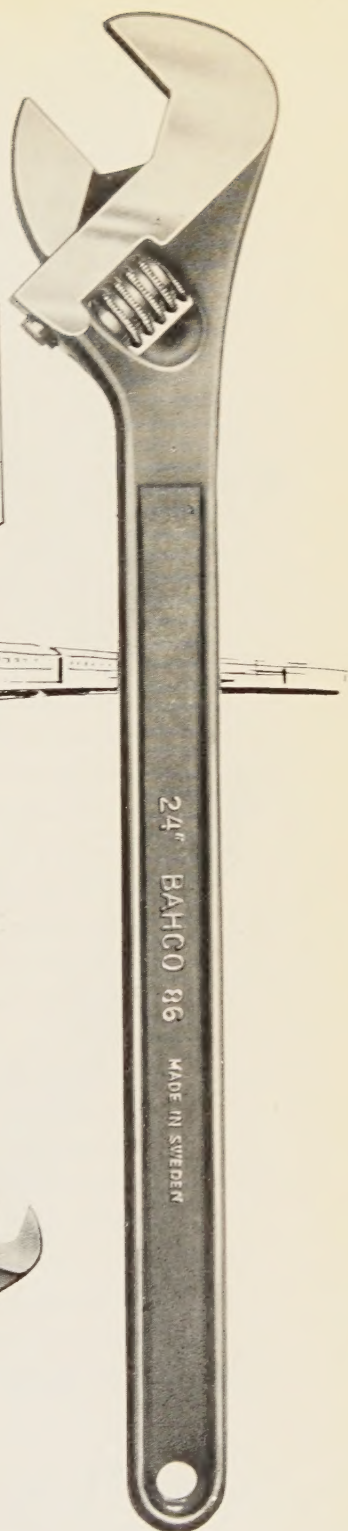
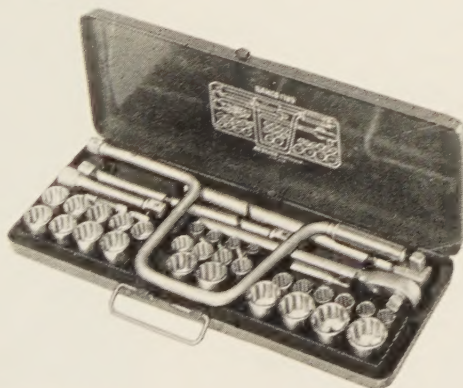
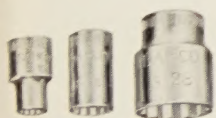
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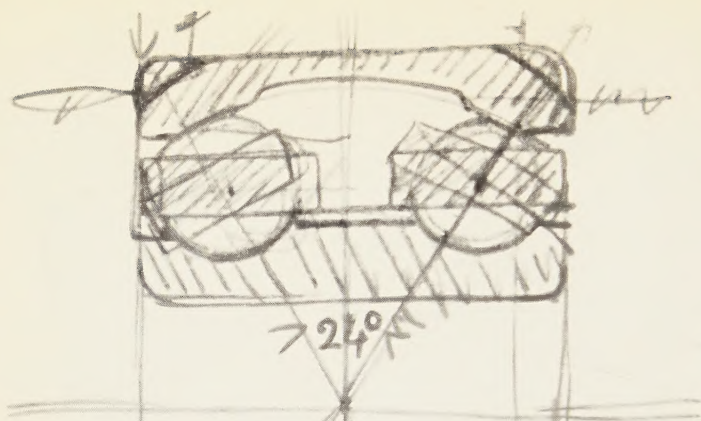
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BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS

ASSOCIATION
(ENGLISH EDITION)

[625 .233]

Replacing the train lighting dynamo by an alternator,

by E. MEYER,

Ingénieur principal au Service du Matériel et des Achats de la Société Nationale des Chemins de fer belges.

1. Present position.

Most of the modern coaching stock is at present equipped with a dynamo, a voltage regulator, and an accumulator battery.

The accumulator battery, which is on charge when the vehicle is running, supplies the train lighting energy when the vehicle is at rest as the dynamo can only be operated by the axle.

There are other, simpler devices but these do not offer the same flexibility in operation.

When the coaches were first equipped with electric lighting, the locomotives were fitted with large dynamos supplying the electric energy for the whole of the train.

This necessitated the provision of cables and cable connections with plugs and terminals on all coaches.

When the coaches were stationary without locomotive, the lighting current supply was ensured by an accumulator battery installed in the luggage van.

This system, though simple and inexpensive, is not of much use to the operator. A failure of the dynamo, the battery or any coupling connection will automatically and immediately cause the failure of the lighting throughout the train.

A different system has been developed by the Swedish State Railways, and has, to the author's knowledge, not been used on any other system.

The Swedish railways are almost completely electrified and, on the few diesel-operated lines, the train heating is obtained by means of an alternator installed on the locomotive.

In all these cases, the electric heating system is fed by a 1 000 V, 16 $\frac{2}{3}$ cycles supply. This system can also be used to feed the lighting system. In each coach, a small transformer and rectifier are connected to the heating circuit which feed the low-tension installation of the vehicle.

The equipment of each coach is completed by an accumulator battery.

This system is particularly attractive in Sweden as the lighting completely ceases to be dependent on the driving of a dynamo by an axle, a method of operation specially unreliable in a country where snow must be reckoned with for many months of the year.

Unfortunately, this solution can only be adopted in countries which are completely, or almost completely, electrified on the A.C. system. Damage to the heating cables will not cause the failure of the train lighting as the battery in each coach can supply the lighting current for the rest of the trip.

Another solution has been adopted by the Italian Railways. Each vehicle is equipped with a large accumulator battery which is charged during stops by means of powerful charging sets.

The capacity of the battery must be such as to ensure the lighting of the vehicle during the whole night trip.

In this way, it is possible to avoid the installation of dynamos and voltage regulators and to eliminate the problem of driving the dynamo, but a complex and costly servicing organisation is required for the charging of the batteries.

In practice, two batteries must be kept for each coach: one in the coach, the other one on charge, and these must be changed over after each trip.

This system is hardly commendable, and the author believes that the arrangement will not be extended further.

2. Drawbacks of the dynamo-regulator system.

The dynamo, mechanically linked with the axle, is a machine with varying speed. It must therefore be associated with a

voltage regulator covering the whole speed range of the machine.

Moreover, because of the charging of the accumulator battery, the direction of the voltage supplied by the dynamo must be independent of the running direction of the vehicle. The dynamo must therefore be equipped with a brush shifting device, usually with friction, actuated by the direction of running.

Although the dynamo and voltage regulator system has already been improved in many respects, the number of lighting failures is still too great, and the organisation of a maintenance service designed to reduce the number of failures is expensive, as it is necessary to provide special staff at all important stations of the railway system.

In view of the fact that most of the failures are due to imperfect working of the dynamo, especially in the part of the equipment designed to ensure the shifting of the brushes, and to the delicate working of a voltage regulator exposed to numerous shocks, the author suggested to the Belgian S.E.M. company that they should study an arrangement based on an entirely different principle.

With this arrangement, the dynamo would be replaced by an alternator, associated with a rectifier. The latter would feed the battery and the lighting installation.

3. Principle of the new device.

An alternator driven at variable speed but with constant excitation supplies an A.C. voltage whose no-load voltage and frequency are proportionate to the speed.

The voltage drop in the alternator increases with the frequency of the cur-

rent, i.e. with the rotating speed of the machine.

The same applies to the value of the impedance against which the alternator will deliver current as long as this

the alternator delivering against a complex impedance is constant.

The no-load voltage increases in proportion to the speed.

The voltage drop in the armature and

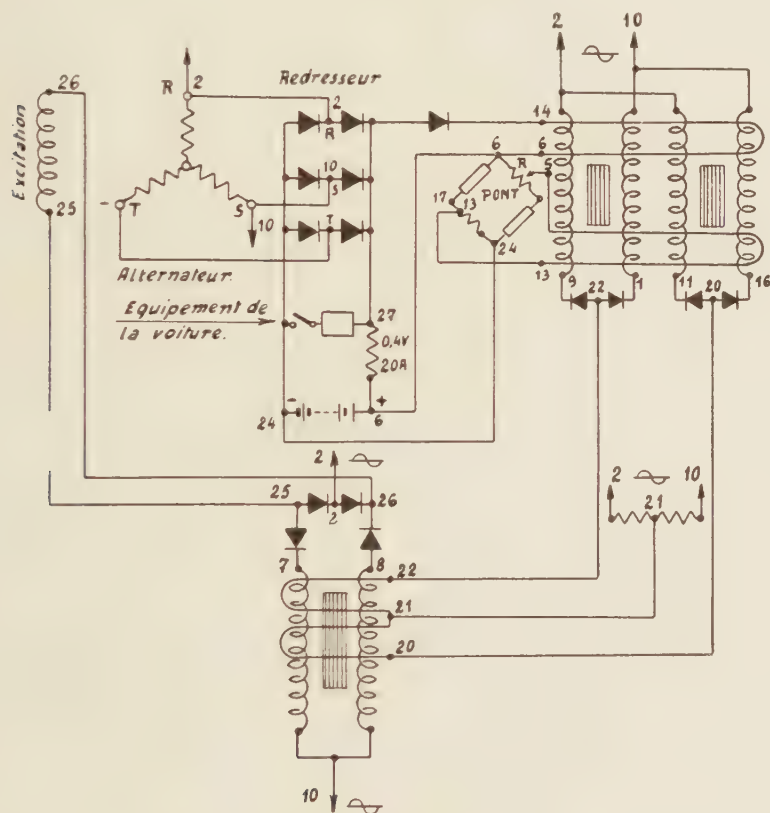


Fig. 1. — Regulation diagram of the alternator used as generator for the train lighting.

N. B. — Redresseur = rectifier. — Alternateur = alternator. — Excitation = excitation. — Equipement de la voiture = vehicle equipment.

impedance is predominantly a self-impedance.

The current delivered will depend on the interplay of the different factors with opposite effects.

Let us assume that the excitation of

the armature reaction increase in proportion to the speed if one ensures that the self-impedance portion of the armature reaction is preponderant.

It would therefore be possible to arrange for the current supplied by the

machine to be independent of the rotating speed provided that the alternator characteristics and particularly the value of the armature reaction are suitably chosen.

The current supplied by the alternator will be rectified in a static rectifier, which brings us back to the case of the dynamo and its regulator, but without the complexity of that device.

4. Practical solution.

The solution outlined above is still too sketchy to be ready for practical realisation.

The S.E.M. company has developed a device which adds to the advantages of the theoretical solution some new and very valuable features (fig. 1).

For the alternator must not only supply the alternating current which, when rectified, will charge the accumulator battery, but it must also, if possible, modulate the intensity of this current as a function of the battery charge or, which amounts to the same thing, modulate the voltage of the rectifier as a function of the battery voltage.

This result has been obtained through replacing the constant excitation by a variable excitation which is a function of the following factors :

- (a) the battery voltage,
- (b) the voltage of the lighting system.

The excitation of the alternator will therefore be modulated as a function of these two factors.

There could be no question of obtaining this effect by means of a normal voltage regulator as this is the very device it was intended to eliminate.

It was therefore necessary to obtain the voltage variation by means of the alternating current.

It is possible to obtain a modulated continuous current in the following way.

The D.C. circuit to be regulated is fed by an alternating voltage. This circuit includes a rectifier. Also inserted in the circuit is a saturable self-inductance coil. The drop of the alternating voltage at the terminals of the coil will depend on the saturation of the latter. The greater the saturation, the smaller will be the voltage drop, and the heavier the current absorbed by the circuit.

This device is adopted to set up the excitation circuit. The excitation current will therefore ultimately be a function of the state of saturation of the inductance coil.

The saturation current of the inductance coil must be regulated in keeping with the purpose which one has in mind.

In practice, the saturation of the inductance coil must be brought about by a continuous current likewise originating from a circuit fed by the alternator, followed by a small rectifier.

The inductance coil can be saturated by the effect of two distinct continuous currents :

- (a) *A continuous current whose intensity and direction will depend on the voltage of the battery terminals, serves as reference current.*

If the voltage at the battery terminals is normal, the continuous current will be non-existent. This effect is obtained by feeding, by means of the voltage to be controlled, a bridge whose two opposite

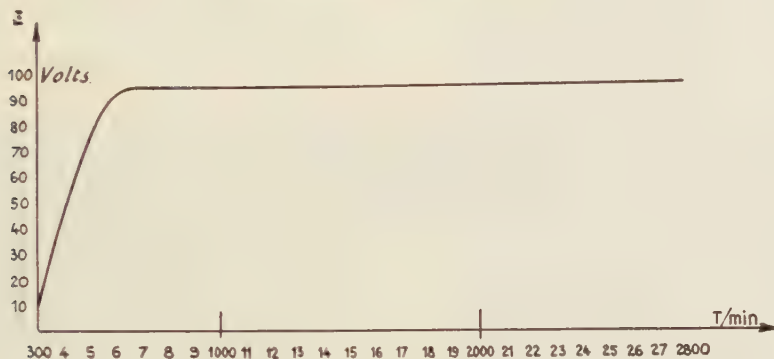
No-load characteristic.*No-load voltage measured as a function of speed.*

Fig. 2.

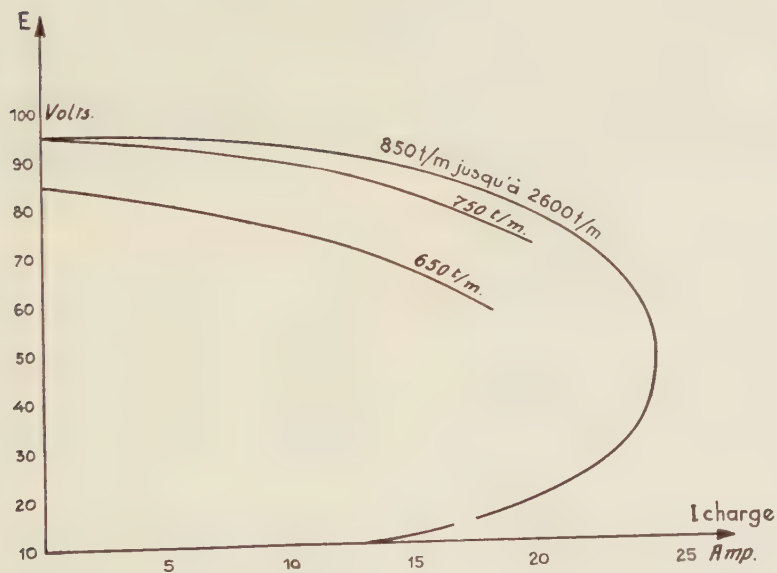
Load characteristic.*Voltage supplied by the rectifier as a function of the battery charging current.*

Fig. 3.

branches are fitted with non-linear resistances.

The control current will then be positive or negative according to the value of the voltage controlled.

The effects of these two currents are added to or subtracted from each other, according to the direction of the first current.

The excitation current will thus be

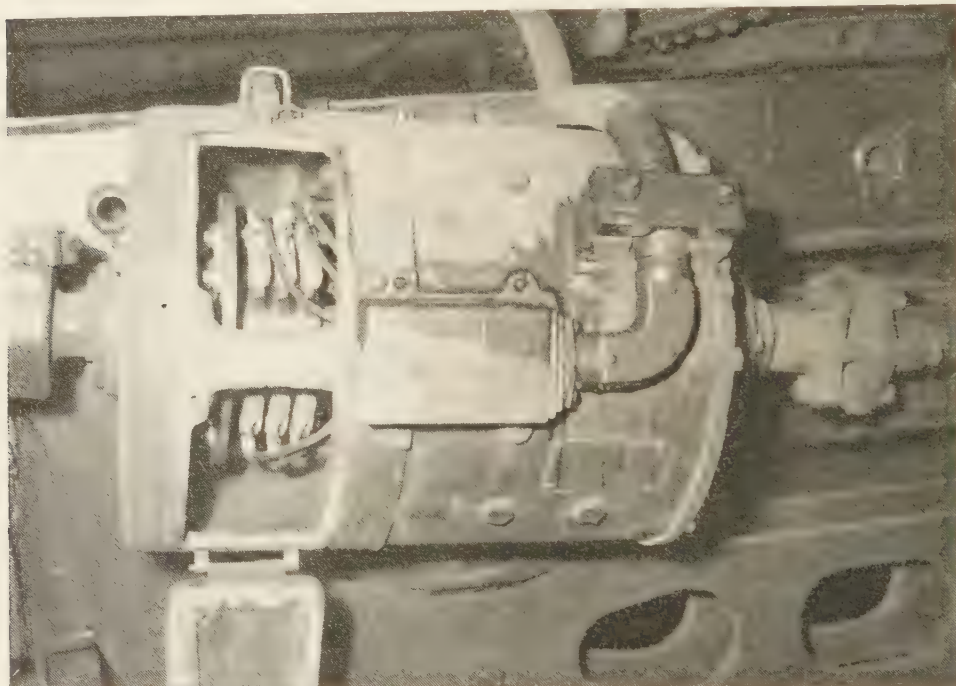


Fig. 4. — Alternator mounted on the bogie frame.

(b) *A continuous current whose intensity (but not direction) will depend on the battery charging current.*

This current is obtained by connecting a fixed resistance in the battery charging circuit. The voltage drop produced in this resistance is used to feed a circuit with an element of non-linear resistance.

This circuit also serves to ensure the saturation of the inductance coil used to regulate the excitation current.

statically modulated by the battery voltage and the battery charging current.

The current for the lighting circuit has no bearing on the voltage regulation, as this current does not flow through the resistance series-connected with the battery.

The trials which have been carried out have shown that :

(1) all other things being equal, the continuous voltage is absolutely indepen-

dent of the speed, from a speed of 700 r.p.m. upwards (fig. 2):

(2) the continuous voltage is absolutely independent of the current absorbed by the lighting system;

(3) the continuous voltage follows a declining curve, as a function of the growing intensity of the current absorbed by the battery (fig. 3).

This curve is identical to the one obtained from a shunt dynamo at constant speed if the voltage at the terminals is raised as a function of the current supplied.

The two first results are obtained due to the adoption of the basic principle. The third result is obtained due to the regulating device.

These trials have been so satisfactory that it was decided to install the apparatus in a prototype vehicle which was equipped during the last three months of 1955.

The equipment consists of an alternator, mounted and driven by cardan shafts in the same way as the dynamos in the latest 200 vehicles of the Belgian National Railways (fig. 4).

The rectifier, enclosed in a receptacle containing the cell cooling oil, is slung below the underframe.

The control gear assembly is mounted in the place usually reserved for the voltage regulator (fig. 5).

After twelve months of normal service, the apparatus was still in excellent working order.

5. Future perspectives for the new device.

When it was decided, during the second half of 1956, to construct a series

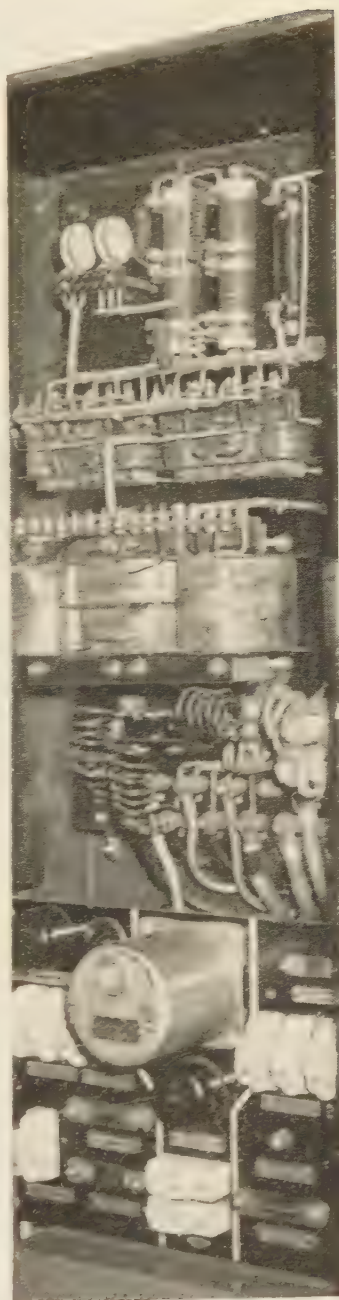


FIG. 5 — Control gear assembly (typical installation)

of new coaches, it was agreed to equip 35 of these vehicles with the new device.

The advantages which the railway operator can obtain from this new apparatus are these :

(a) Reduction of the number of failures due to the elimination of the brush shifting device, the dynamo commutator and the usual voltage regulator;

(b) A fairly considerable reduction in maintenance costs.

This saving is of particular importance as the Belgian Railways find it more and more difficult to recruit the number of trained electricians required for electric maintenance work.

These difficulties are due to two reasons :

(a) The extension of railway electrification which calls for a greater number

of highly qualified staff for the maintenance of sub-stations, catenaries and rolling stock;

(b) The continuous spreading of the use of electricity in the industrial sphere.

It is only with great difficulty, if at all, that the technical colleges in Belgium succeed in training the ever increasing number of qualified electricians required by industry. It is therefore the duty of the Belgian National Railways, by rationalising the duties and improving the equipment, to save qualified man-power which is in danger of being in short supply.

If, as there is every reason to expect, the results eventually obtained on these 35 coaches are as favourable as those obtained on the prototype vehicle, it is therefore intended to standardize this device for all future rolling stock.

Diagrams to determine the dimensions of a locomotive boiler,

by Siegfried BÜTTNER,

Institute for railway vehicles, Berlin-Adlershof.

(*Deutsche Eisenbahntechnik*, March 1956.)

In order to simplify and reduce the mathematical effort for the calculation and design of steam locomotives, diagrams have been prepared which permit the determination of the dimensions of a locomotive boiler and the calculation of the heat transmission of the boiler. These diagrams

are usually different from the actual values as the designer is, in many cases, obliged to rely on arbitrary assumptions. The diagrammatic method thus represents a marked simplification of mathematical effort while, at the same time, affording a better appreciation of the quantitative effect of the

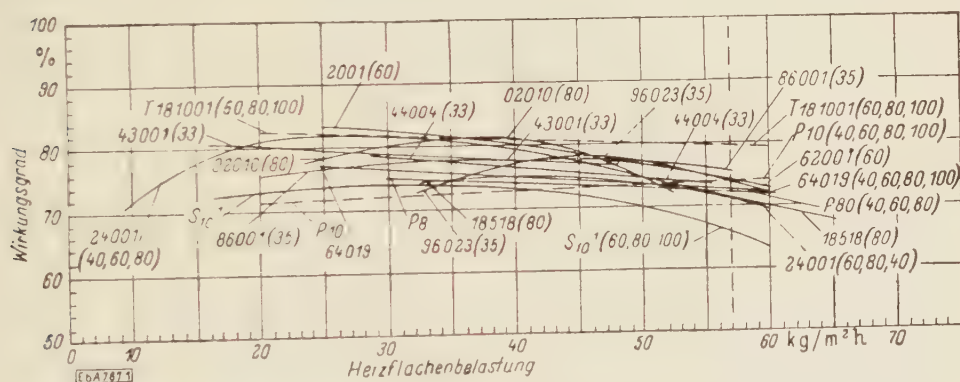


Fig. 1. — Boiler efficiencies, according to running tests. (The figures in brackets represent speeds in km/h.)

N. B. — Wirkungsgrad = efficiency. — Heizflächenbelastung = heat transferred per unit of surface.

are based on the best known and latest formulas and characteristics. They contain, in diagrammatic form, some of the data required for the calculation of a locomotive boiler. The accuracy of a nomogram is of course limited by the scale that can be adopted in view of the restricted space available. But this accuracy is sufficient for all practical applications. Also in a numerical calculation, the results are nor-

modification of a dimension or a calculation factor. The application of each of these diagrams is explained by means of a concrete example.

1. Hourly steam output of the boiler (Diagram C 1).

The requisite size of the heating surface and the requisite hourly steam output

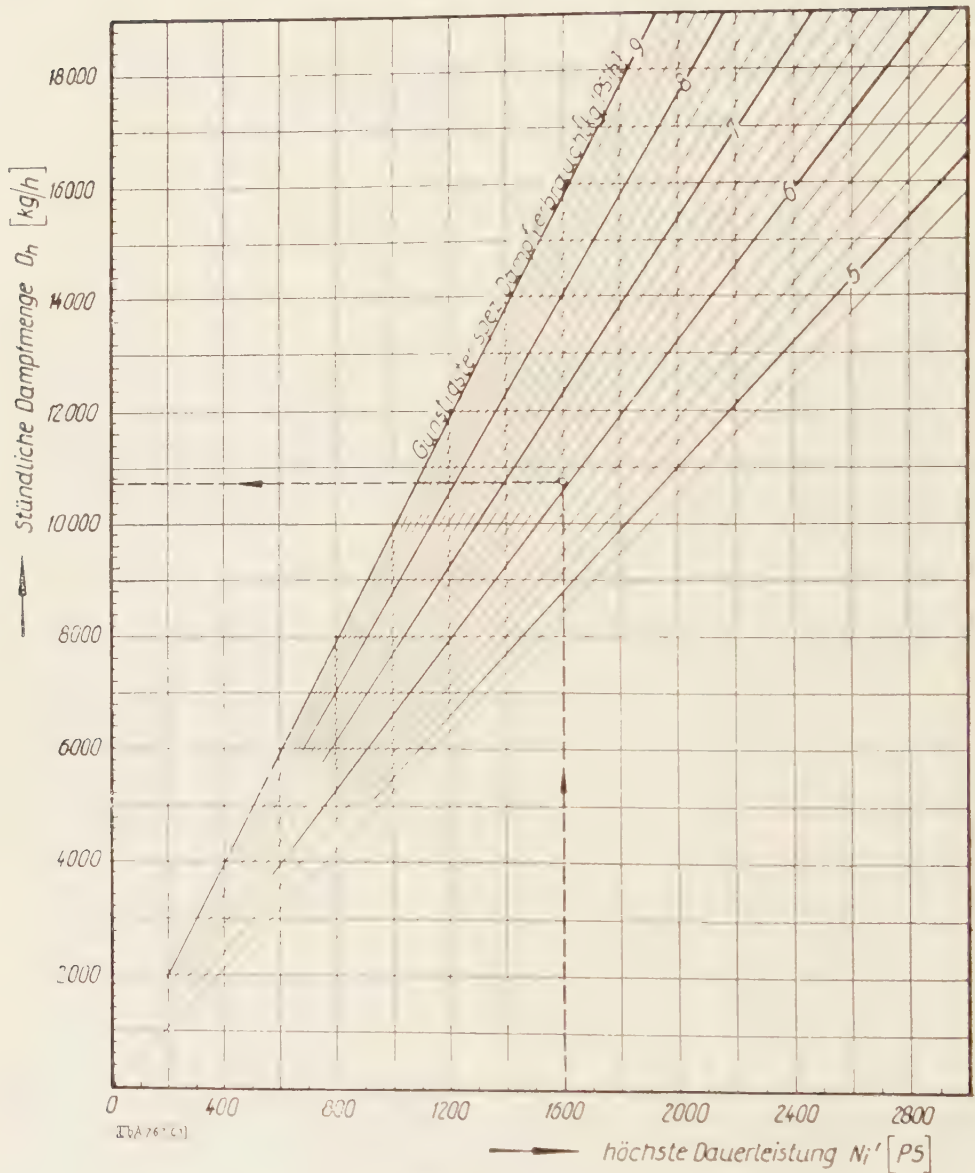


Diagram C 1.

Requisite boiler steam output.

Example : Maximum indicated locomotive power : $N_i = 1\,600$ HP; specific consumption : $d_i = 6.1$ kg per indicated HP/h. — One obtains, for the requisite hourly steam output : $D_h = 10\,750$ kg/h.

N. B. — Abscissae = maximum indicated power. — Ordinates = hourly steam output. — Günstigster spez. Dampfverbrauch = optimum specific steam consumption.

depend, first of all, on the indicated horsepower :

$$N_i = \frac{Z_i V}{270} \text{ HP) } \quad (1)$$

and on the expected specific steam consumption d'_i . If the additional steam consumption for brake, train heating and boiler feed is taken into account by means of a factor c , the requisite hourly steam output of the boiler amounts to :

$$D_h = c N'_i d'_i \text{ (kg h)} \quad (2)$$

The factor c is generally assumed to be 1.1. The optimum specific steam consumption d'_i can be determined with the aid of ACHTERBERG's formula [2], from Diagram C 2 whilst the indicated power N_i is known from equation (1). In Diagram C 1, the requisite hourly steam output is plotted in accordance with equation (2).

2. Optimum specific steam consumption d'_i (Diagram C 2).

There exist, in the technical literature, several formulae for the calculation of the optimum specific steam consumption. The first calculation was carried out by ACHTERBERG [2] who used the following formula :

$$d' = \left(a - \frac{t_u}{b}\right) \left(1.22 - \frac{d}{215}\right) \text{ (kg per indicated HP/h)} \quad (3)$$

where a and b are constants dependent on the boiler pressure, t_u the temperature of the superheated steam, and d the cylinder diameter in centimetres. As the constants quoted by ACHTERBERG could only be applied for boiler pressures up to 16 kg/cm², they were recalculated by MEINEKE [3] on the strength of locomotive tests, for pressures up to 20 kg/cm². According to him, the equation should be written :

$$d' = \left(a - \frac{t_u}{b}\right) \left(1.24 - \frac{d}{250}\right) \text{ (kg per indicated HP/h)} \quad (4)$$

with the constants a and b shown in Table 1.

TABLE 1. — Constants a and b .

p in kgs.pr.sq.cm. (effective)	12	14	16	18	20
a	13	12.4	11.8	11.3	10.8
b	59	64	69	74	80

Equation (4) applies to locomotives with superheated steam and single-stage steam expansion, and only between the entropy limits of $s = 1.68$ to 1.72 of the intake steam. For saturated steam, one may use HELMHOLTZ's formula :

$$d' = \left(0.675 + \frac{1.9}{3\sqrt{I}}\right) \frac{8.95 + 66 p^{-1.5}}{1 + 0.003 (1 + 0.01 p) (t_u - t_s)} \text{ (kg per indicated HP/h)} \quad (6)$$

where I = cylinder volume in dm³,

p = steam pressure in kg/cm²,

t_u = temperature of the overheated steam in °C,

$$d' = 13.4 - \frac{d}{31.5} \quad (5)$$

POSTUPALSKY [4] has recently published an amplified formula valid for all steam conditions :

t_s = temperature of the steam in the steam chest.

But, because of the assumptions that must be made, even this formula can only be regarded as an approximation so that there

$$d_i \cdot \left(\frac{1}{b} - \frac{t_u}{b} \right) \left(e - \frac{d_i \cdot l}{f} \right) \left[\text{kg/PS.h} \right]$$

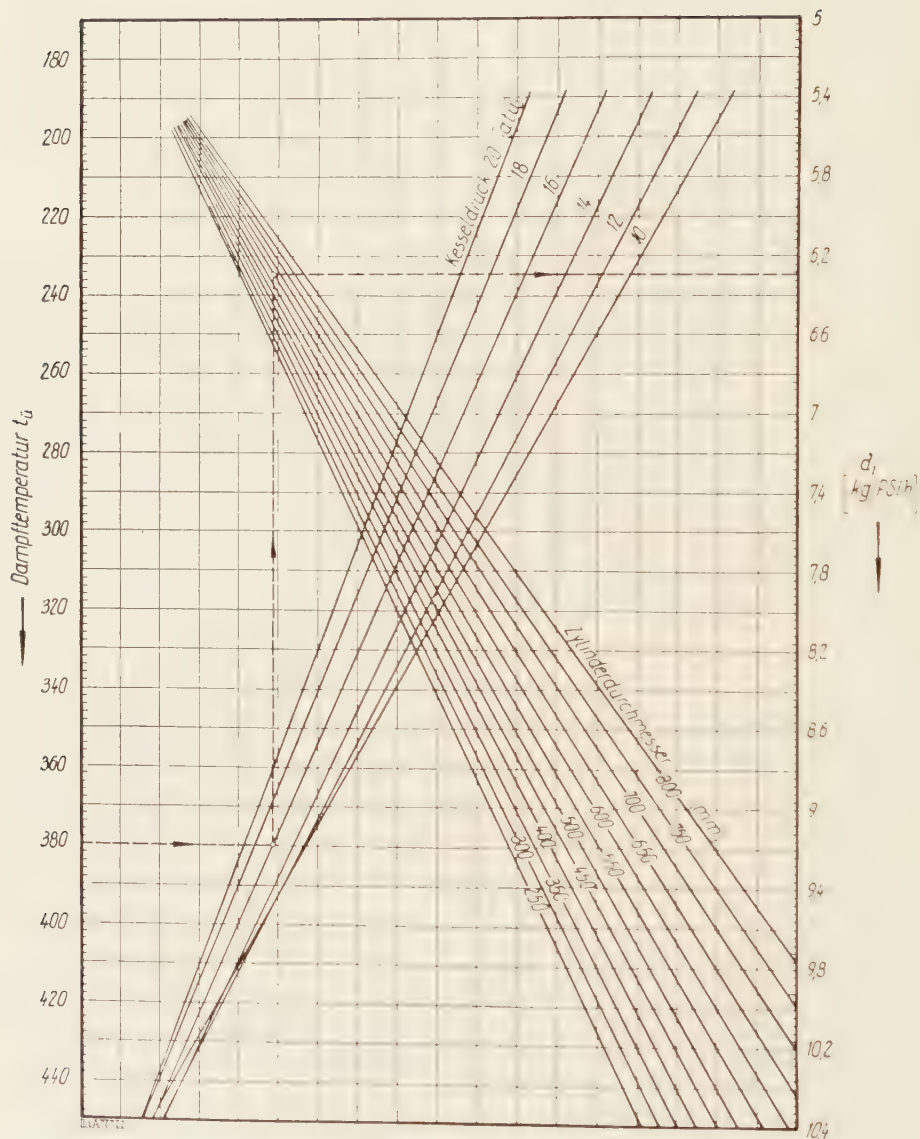


Diagram C 2.

Optimum specific steam consumption.

Example : Steam temperature : 380° C; boiler pressure : 16 kg/cm² (effective); cylinder diameter : 600 mm.

From the diagram, one obtains the optimum specific steam consumption : $d_i = 6.29$ kg per indicated HP.h.

N. B. — Dampftemperatur = steam temperature. — Kesseldruck = boiler pressure. — Zylinderdurchmesser = cylinder diameter.

is no reason to abandon ACHTERBERG's simple formula, an opinion which is also shared by NORDMANN [5]. There also exist equations, developed by SYROMJATNIKOFF

[6], for the specific steam consumption as a function of the intake, but these are even more complicated. SYROMJATNIKOFF suggests, e.g. for :

$$\left. \begin{aligned} \varepsilon = 0.1; d_i &= \frac{2700}{100 - \frac{62.6 - 0.157 \Delta t}{\sqrt{n}}} \cdot \frac{0.2 y_{sch} - 0.4948 y_{ab}}{4.635 - 0.314 n} (0.77 + 2.3 \varepsilon_o) \\ \varepsilon = 0.2; d_i &= \frac{2700}{100 - \frac{44.3 - 0.0111 \Delta t}{\sqrt{n}}} \cdot \frac{0.3 y_{sch} - 0.415 y_{ab}}{6.835 - 0.278 n} (0.79 + 2.1 \varepsilon_o) \\ \varepsilon = 0.3; d_i &= \frac{2700}{100 - \frac{36.15 - 0.090 \Delta t}{\sqrt{n}}} \cdot \frac{0.4 y_{sch} - 0.354 y_{ab}}{8.557 - 0.166 n} (0.81 + 1.9 \varepsilon_o) \\ \varepsilon = 0.4; d_i &= \frac{2700}{100 - \frac{31.31 - 0.078 \Delta t}{\sqrt{n}}} \cdot \frac{0.5 y_{sch} - 0.304 y_{ab}}{9.93} (0.84 + 1.6 \varepsilon_o) \\ \varepsilon = 0.5; d_i &= \frac{2700}{100 - \frac{28.00 - 0.070 \Delta t}{\sqrt{n}}} \cdot \frac{0.6 y_{sch} - 0.258 y_{ab}}{11.90} (0.86 + 1.4 \varepsilon_o) \\ \varepsilon = 0.6; d_i &= \frac{2700}{100 - \frac{25.56 - 0.064 \Delta t}{\sqrt{n}}} \cdot \frac{0.7 y_{sch} - 0.223 y_{ab}}{11.90} (0.89 + 1.1 \varepsilon_o) \end{aligned} \right\} \quad (7)$$

where n = number of revolutions.

y_{sch} = density of the steam in the steam chest,

y_{ab} = density of the steam in the exhaust,

ε_o = proportion of cylinder clearance,

ε = filling ratio.

In order to determine the optimum specific steam consumption according to SYROMJATNIKOFF, the optimum speed of the locomotive must be taken into account from which the intake can, according to LUBIMOFF [7], be worked out with the aid of the formula :

$$\varepsilon = \frac{1}{4} [6.45 - \sqrt{41.6 - 8 \alpha (n + 3)}] \quad (8)$$

TABLE 2. — Optimum specific steam consumption.

Formula	(4)	(6)	(7)
Specific steam consumption (kgs. pr. indicated HP pr. hr.)	6.4	6.35	6.55

Table 2 shows the optimum specific steam consumption in accordance with the three equations quoted above, and for identical conditions. As an example, we have taken a passenger locomotive P C P, series 23, with the following characteristics :

Cylinder diameter	550 mm
Piston stroke	660 mm
Steam pressure	16 kg/cm ² (effective)
Steam superheating temperature	380 °C
Optimum speed	84.6 km/h
Intake, according to (8) . . .	30 %
Driving wheel diameter . . .	1 750 mm
Tractive effort index . . .	0.247
Ratio of cylinder clearance .	0.09

The comparison shows that ACHTERBERG's simple formula yields a suitable mean value for the locomotive design. Diagram C 2 is therefore based on equation (3).

3. Hourly fuel consumption B_h (Diagram C 3).

If the hourly steam output D_h is known, the hourly fuel consumption of a locomotive can be calculated from the equation :

$$B_h = \frac{D_h (i'' - t_w)}{\eta_k H_u} \text{ (kg h)} \quad (9)$$

where i'' = heat contents of the superheated steam, in thermal units per kilogram,

t_w = temperature of feed water in °C,

H_u = lower calorific value of the fuel in thermal units per kilogram, and

η_k = boiler efficiency.

According to the results of tests carried out by NORDMANN [8] which are shown in figure 1, the boiler efficiency η_k can be assumed to be $\eta_k = 0.68$ if the feed water temperature is about 20 °C, and $\eta_k = 0.72$ if the feed water is preheated to 90 °C.

4. Boiler unit stress A (Diagram C 4).

The notion „grate unit stress” or „boiler unit stress” :

$$A = \frac{B_h H_u}{R \cdot 10^6} \quad (10)$$

introduced by STRAHL [9] has been rendered applicable to all types of fuel by MEINEKE [10] due to the introduction of a fictitious grate surface ρR . The boiler unit stress thus amounts to :

$$A = \frac{B_h H_u}{\rho R \cdot 10^6} \quad (11)$$

This equation has been used for the plotting of Diagram C 4, with the following values :

$\rho = 1.1$ for coke and large-size coal.

$\rho = 1.0$ for ordinary coal.

$\rho = 0.91$ for lignite from Central Germany.

$\rho = 0.83$ for hard lignite from Bohemia,

$\rho = 0.8$ for wood.

$\rho = 0.5$ for anthracite and coal dust.

This diagram not only permits the determination of the boiler unit stress A but also, if the latter is known, the determination of the requisite quantity of fuel per hour, B_h .

5. Tube index K (Diagram C 5).

In locomotive design, the „tube index” is defined as :

$$K = h Q \quad (12)$$

where h = heating surface of the tube in contact with the combustion gases, and

Q = cross-sectional area of tube available for the combustion gases.

For the DIN standard tubes normally used in Germany, the cross-sectional areas and indices of the tubes are given in a numerical table in the Henschel Manual

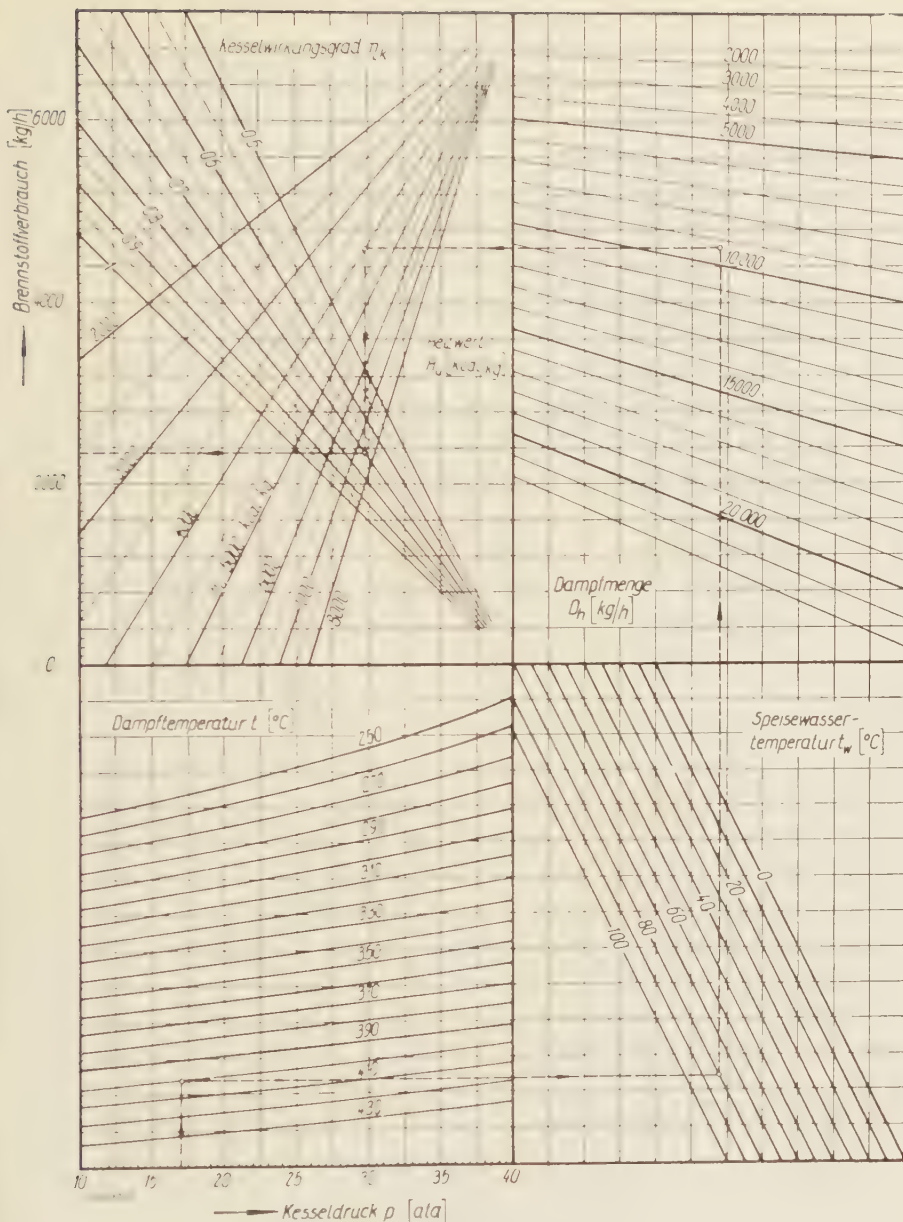


Diagram C 3.

Hourly fuel consumption.

Example : Boiler pressure : $p = 16 \text{ kg/cm}^2$ (effective) i.e. 17 kg/cm^2 (absolute) approximately;
 Steam temperature : $t = 400^\circ \text{C}$;
 Feed water temperature : $t_w = 80^\circ \text{C}$;
 Hourly steam output : $D_h = 9350 \text{ kg/h}$;

Calorific value of the coal : $H_u = 4000$ thermal units per kilogram;
 Boiler efficiency : $\eta_k = 0.7$;
 One obtains an hourly fuel consumption of :
 $B_h = 2350 \text{ kg/h}$.

N. B. — Kesseldruck = boiler pressure. — Brennstoffverbrauch = fuel consumption. — Kesselwirkungsgrad = boiler efficiency. — Heizwert = calorific value. — Dampfmenge = quantity of steam. — Dampftemperatur = steam temperature. — Speisewassertemperatur = feed water temperature.

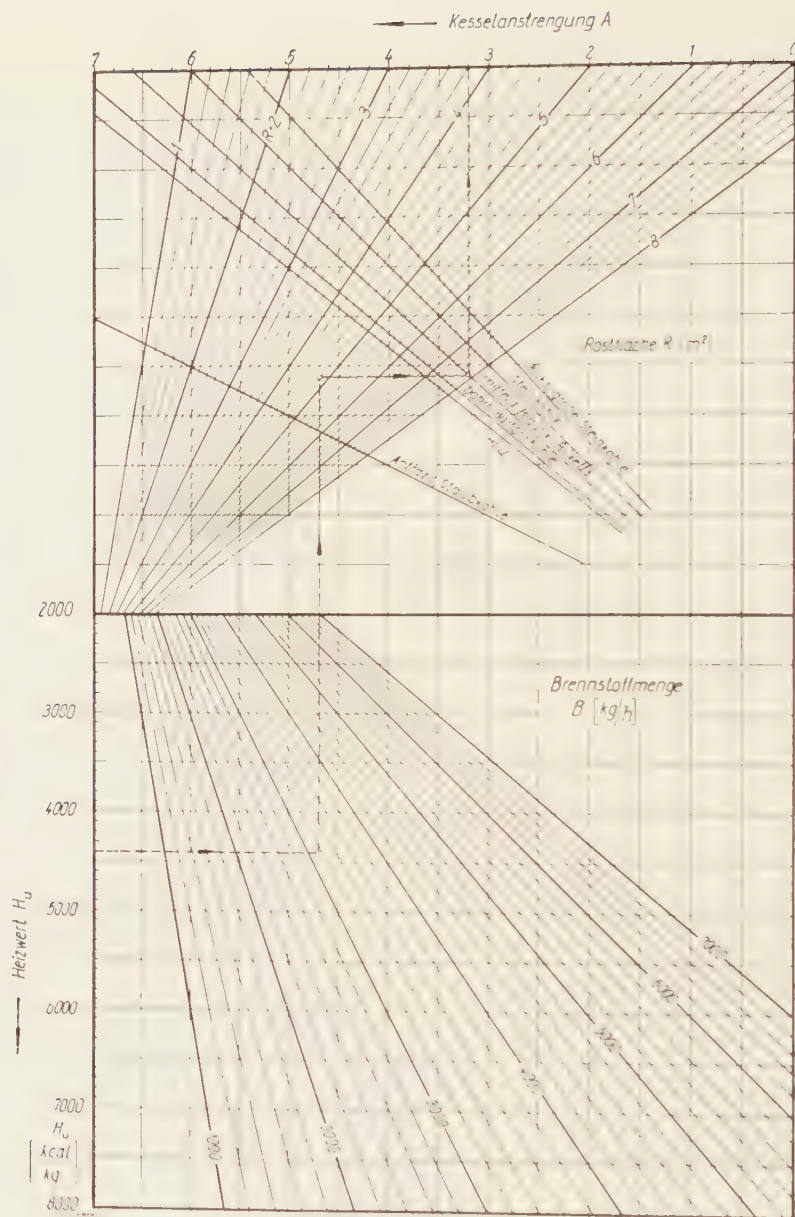


Diagram C 4.

Boiler unit stress A (according to STRAHL).

Example : Lignite briquettes;

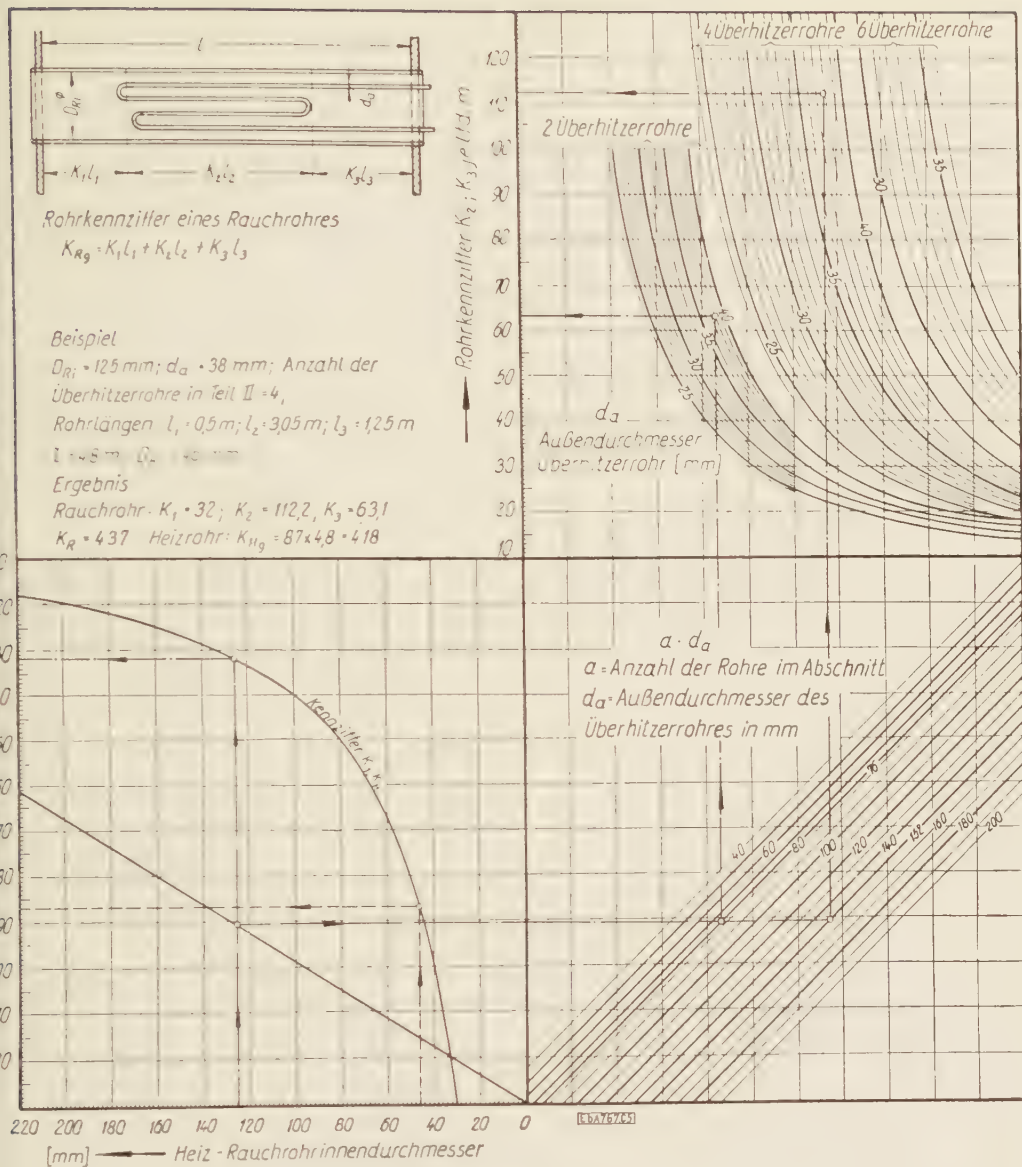
Calorific value : $H_u = 4\,400$ thermal units per kilogram;

Quantity of fuel : $B = 3\,180$ kg h;

Grate surface : $R = 4.8$ m²;

One obtains : $A = 3.2$.

N. B. — Rostfläche = grate surface. — Heizwert = calorific value. — Brennstoffmenge = quantity of fuel. — Kesselanstrengung = boiler unit stress. — Koks, grobe Steinkohle = coke, large-size coal. — Steinkohle = coal. — Mitteld. Braunk-Briketts = lignite briquettes from Central Germany. — Böhm. Hartbr.-Kohle = hard lignite from Bohemia — Holz = wood. — Anthrazit, Staubkohle = anthracite, coal dust.



Example :

$D_{Ri} = 125 \text{ mm}$; $d_a = 38 \text{ mm}$;

Number of superheater tubes in part II = 4;

Tube lengths : $l_1 = 0,5 \text{ m}$; $l_2 = 3,05 \text{ m}$;

$l_3 = 1,25 \text{ m}$; $l = 4,8 \text{ m}$; $D_{Hi} = 46 \text{ mm}$.

Result :

Smoke tube : $K_1 = 31$; $K_2 = 112,2$; $K_3 = 63,1$;

$K_R = 437$. Fire tube : $K_{Hg} = 87 \times 4,8 = 418$.

N. B. — Rohrkennziffer $K_1 \dots$ = tube index pr. linear meter. — Überhitzerrohre = superheater tubes. — Außendurchmesser Überhitzerrohr = external diameter of superheater tube. — a = number of tubes in the cross section. — d_a = external diameter of the superheater tube in millimetres. — Rohrkennziffer eines Rauchrohres = tube index of a smoke tube. — Rauchvolumendurchmesser = inner diameter of smoke tubes.

Cross-sectional areas and indices of DIN tubes.

Fire tubes and non-occupied smoke tubes			Superheater						Tube index K pr. linear m		
Code according to DIN 32 050	Heating surface <i>fb</i> in sq. m per linear meter	Cross-sectional area available to the gases, in sq. cm	Superheater tubes in Part II (wholly occupied)		Cross-sectional area of smoke tube available to gases, in sq. cm		External heating surface of superheater pr. linear		Fire tube or non-occupied smoke tube (Part I)	Smoke tube	
			Number	Description	Part II (wholly occupied)	Part III (half occupied with two superheater tubes)	Part II (wholly occupied) m ²	Part III (half-occupied) m ²		Wholly occupied, Part II	Half-occupied, Part III
38 × 2.5	0.1038	8.55							121.1		
44.5 × 2.5	0.124	12.25							101.1		
51 × 2.5	0.1445	16.62							87.0		
54 × 2.5	0.154	18.85							81.7		
60 × 3	0.1698	22.9							74.1		
63.5 × 3	0.1807	26.0							69.6		
70 × 2.5	0.204	33.2							61.5		
76 × 3	0.22	38.5							57.17		
79 × 3	0.2294	41.9							54.8		
108 × 3.75	0.316	79.4	4	25 × 3	59.7	69.5	0.3144	0.157	39.8	105.6	68.0
118 × 4	0.3455	95.0	4	30 × 3.5	66.72	80.86	0.3772	0.1886	36.4	108.4	66.0
133 × 4	0.393	122.7	4	35 × 4	84.22	103.48	0.44	0.22	32.0	98.9	59.2
			4	38 × 4	77.3	100.0	0.4776	0.2388		112.2	63.1
143 × 4.25	0.4225	142.1	4	38 × 4	96.6	119.3	0.4776	0.2388	29.75	93.1	55.4
152 × 4.25	0.451	162.0	4	38 × 4	116.6	139.3	0.4776	0.2388	27.85	79.6	49.5
171 × 4.5	0.509	206.0	6	30 × 3.5	163.58		0.5658		24.7	65.65	
185 × 5	0.55	240.5	6	30 × 3.5	197.58		0.5658		22.85	56.4	

Diagram C 5.
Tube indices. (continued)

[11]. For the design and calculation of locomotives destined for export, foreign locomotives, and German Reichsbahn locomotives built with tubes imported from Soviet Russia, this table cannot be used because the diameters and cross-sectional areas of the tubes are not identical with those of the DIN tubes. With the aid of Diagram C 5, it is possible to find the characteristics of tubes of all possible dimensions so that diagram is generally applicable.

The index of an entire smoke tube is composed of the indices of the different tube sections :

$$K_{rg} = K_1 \cdot l_1 + K_2 \cdot l_2 + K_3 \cdot l_3 \quad (13)$$

Diagram C 5 yields the tube indices per running metre separately for each tube section, according to the number of superheater tubes there encountered. The index of the entire tube is then obtained by applying equation (13).

6. Temperature of the gases at the fire box flue sheet (Diagram C 6).

The temperature of the gases at the fire box flue sheet is influenced by the grate unit stress, the combustion losses, the air

excess, and the coal combustion temperature. These influences cannot be reliably determined beforehand. The gas temperature is therefore normally determined from empiric formulas based on tests. On the basis of locomotive tests carried out by LOMONOSOFF, SYROMJATNIKOFF [3] has worked out the following formula :

$$t = 670 \left(\frac{B_h H_u}{10^5 H_b} \right)^{0.3} - 100 \left(\frac{B_h H_u}{10^6 H_b} \right)^3 \quad (14)$$

JANUSCH [12] uses for the calculation of the gas temperature at the fire box flue sheet the formula :

$$t = \frac{4\,200 - 600 \lambda}{\xi} \left\{ \frac{\sqrt{\frac{B_h H_u}{H_b}} + 215}{\sqrt{\frac{B_h H_u}{H_b}} + 2\,000} \right\} \quad (15)$$

whilst KLIE and MÜLLER [11] suggest the formula :

$$t = 1\,416 \left(\frac{A R}{H_b} \cdot \eta_f \right)^{0.226} \quad (16)$$

Other engineers also use the following heat equation to determine the gas temperature concerned :

$$(Q_A - C_{pm} (t_2 - t_1)) + H_b K (t_2 - t_g) - \frac{C_g C_w}{C} \left[\left(\frac{T_2}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] H_b + R \frac{C_R C_w}{C} \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] \left(1 - \frac{C'_g}{C} \right) \quad (17)$$

on the assumption that $t_1 = t_2 = t_3$. After determining the radiation coefficients C , the heat transmission coefficient K and the specific heat C_{pm} , one obtains an equation of the fourth order of the form :

$$\left(\frac{T}{100} \right)^4 + k_1 T + k_2 = 0 \quad (18)$$

The notions of equations (14) to (18) have the following significance :

A = grate unit stress,

B_h = hourly fuel consumption,

H_u = lower calorific value of the fuel in thermal units per kilogram,

H_b = fire box heating surface in m^2 ,

λ = air excess coefficient,

ξ = a combustion room coefficient,

η_f = fire efficiency,

c_{pm} = mean specific heat of the gas within the range of fire box temperatures, in thermal units per kg^0C ,

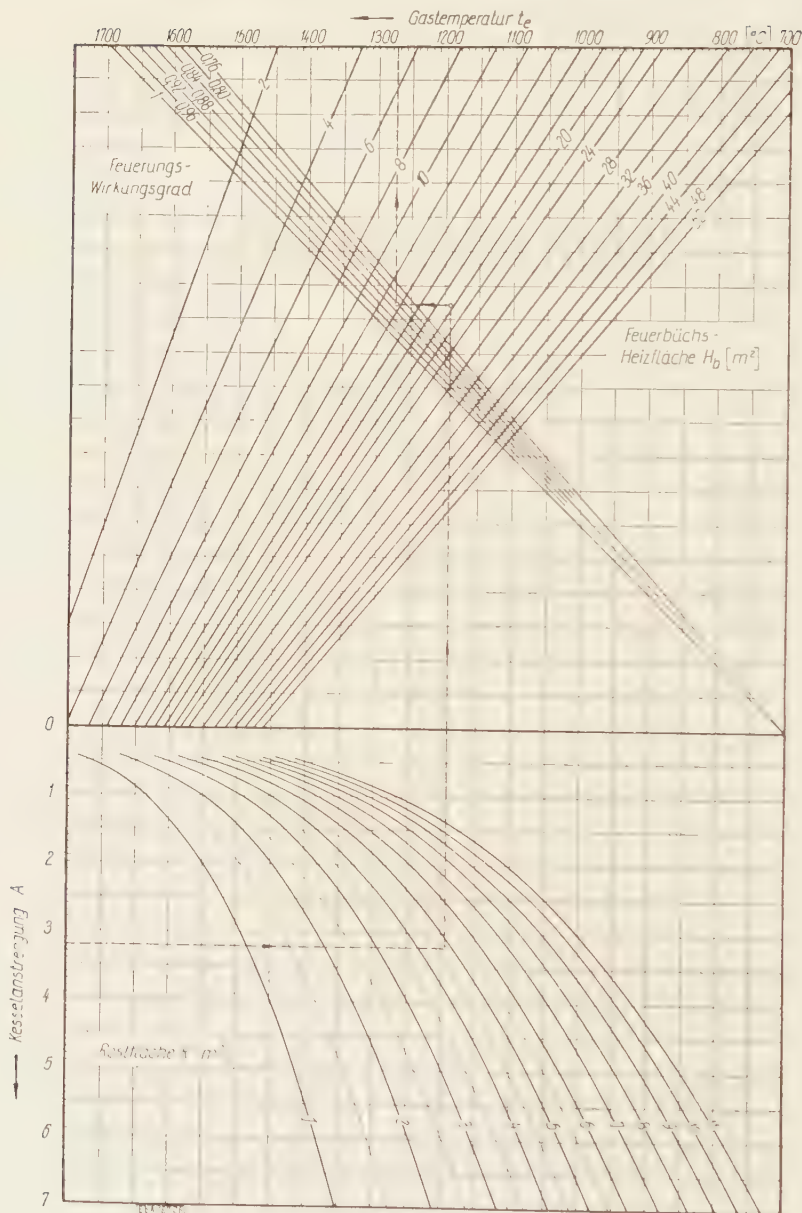


Diagram C 6.
Gas temperature at fire box flue sheet.

Example :

Boiler unit stress : $A \approx 3.2$;

Grate surface : $R = 4.8 \text{ m}^2$;

Fire box heating surface : $H_b = 19.8 \text{ m}^2$;

Fire efficiency : $\eta_f = 0.87$;

One obtains a gas temperature of $t_g = 1275^\circ \text{C}$.

N B. — Feuerungswirkungsgrad = fire efficiency. Feuerbüchsheizfläche = fire box heating surface. — Rostfläche = grate area. — Kesselanstrengung = boiler unit stress.

TABLE 3. — Temperature of the gases at the fire box flue sheet.

Formula	(14)	(15)	(16)	(18)
Gas Temperature in °C	1 180	1 280	1 250	1 230

K = heat transmission coefficient in thermal units per $m^2/h/°C$,

t_s = boiler water temperature,

t = gas temperature, and

C = radiation coefficients in thermal units $m^2 h °C^4$.

Comparing the gas temperatures shown in Table 3, which are calculated from equations (14) to (18) for the following conditions :

$$\begin{array}{ll} B_h = 3\,500 \text{ kg h} & \xi = 0.985 \\ H_u = 4\,000 \text{ thermal} & \eta_f = 0.79 \\ & \text{units per kg} \\ H_b = 19.18 \text{ m}^2 & A = 3.21 \\ \lambda = 1.4 & R = 4.8 \text{ m}^2. \end{array}$$

one finds differences, between equations (15) to (18), of the order of $\pm 4\%$ whilst the temperature calculated from equation (14) differs from that calculated from equation (15) by 8.5% . This comparison thus shows a good agreement between the gas temperature at the fire box flue sheet, and equation (16), formulated by KLIE, yields a reasonable mean value. Diagram C 6 has therefore been based on KLIE's formula.

7. Heat transmission coefficient between the combustion gas and the tube wall (Diagram C 7).

In locomotive design, the heat transmission coefficient between the gases and

the tube wall is generally determined from the equation :

$$\alpha_1 = (3.6 + 0.002 t_m) \frac{w_o^{0.75}}{d^{0.25}} \quad (\text{thermal units } m^2/h/°C) \quad (19)$$

which has been formulated by SCHACK [13] on the basis of numerous tests, and where :

t_m = mean temperature, in centigrades, of the gases in the tube,

w_o = velocity of the gases in m/sec, related to 0 °C and 760 mm mercury column,

d = tube diameter in metres.

The heat transmission coefficient k can be usefully replaced by :

$$k = \beta \alpha_1 \quad (\text{thermal units } m^2/h/°C) \quad (20)$$

where β takes into account the fouling of the tube surface. The fouling factor can be assumed to be $\beta = 0.9$ to 0.95 .

8. Coefficient of heat transmission through gas radiation (Diagram C 8).

In order to be able to calculate the quantity of heat transmitted by the radiation of the gases in the fire box and in the smoke and fire tubes by means of a heat transmission coefficient which takes the radiation of the gases into account, SCHACK [14] has formulated the following equations :

$$\alpha_s \text{ CO}_2 = S \sqrt[3]{p_s} 0.0513 \{ (t_g + t_w) - 30.25 \} \quad (\text{thermal units per } m^2/h/°C) \quad (21)$$

and for the steam :

$$\alpha_s \text{ H}_2\text{O} = S p^{0.85} 0.6 0.107 \{ (t_g + t_w) - 46.5 \} \quad (\text{thermal units per } m^2/h/°C) \quad (22)$$

$$\alpha_1 = (3.6 + 0.002 t_m) \frac{W_0^{0.75}}{d^{0.25}} \quad \text{t}$$

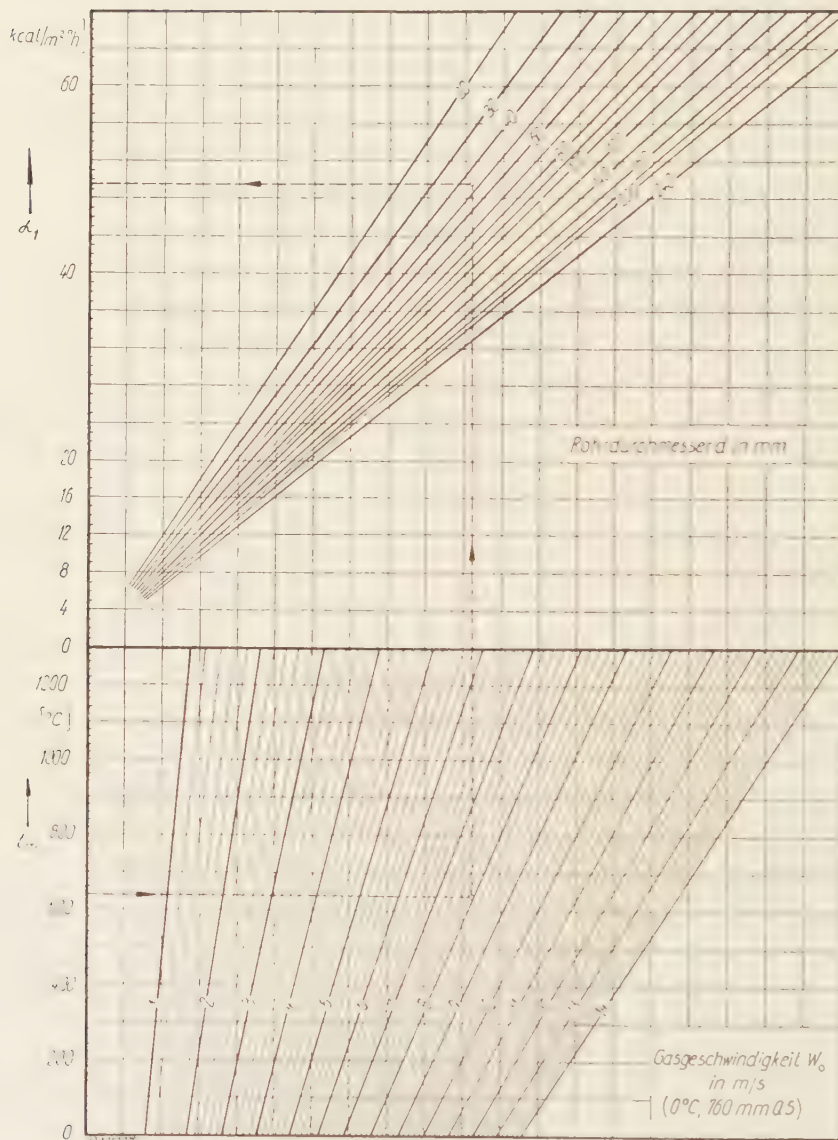


Diagram C 7.
Heat transmission coefficient between combustion gas and tube wall.

Example :

Mean gas temperature : $t_m = 640^\circ \text{C}$;

Tube diameter : $d = 48 \text{ mm}$;

Gas velocity : $W_0 = 7.95 \text{ m/sec}$;

One obtains : $\alpha_1 = 49.6$ thermal units per $\text{m}^2 \text{ h}$.

N. B. — Gasgeschwindigkeit = gas velocity. — Rohrdurchmesser = tube diameter.

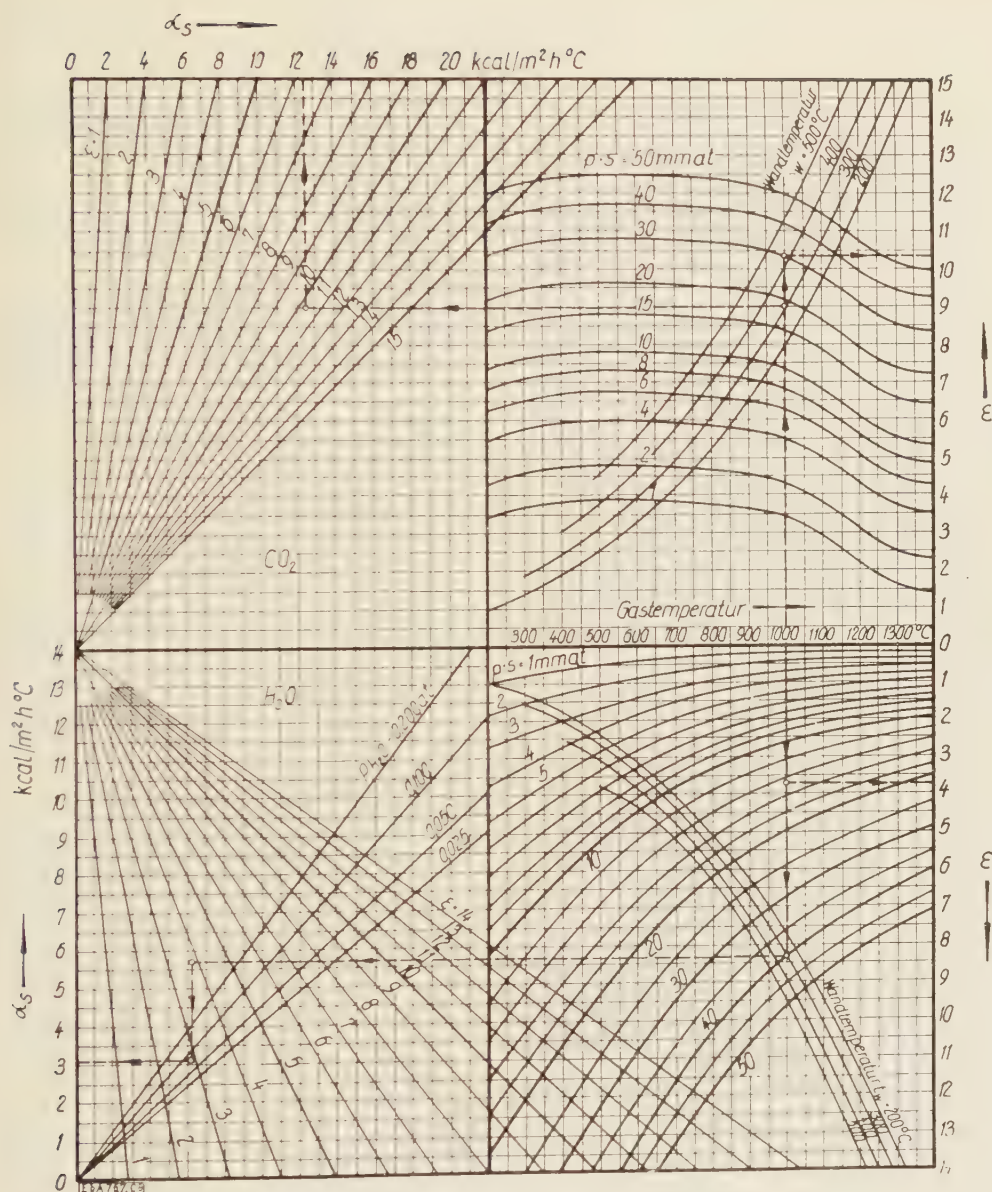


Diagram C 8.
Coefficient of heat transmission through gas radiation.

Example : H₂O.

Given : gas temperature $t_g = 1000^\circ \text{C}$;
 partial pressure $p_{\text{H}_2\text{O}}$;
 thickness of layer s ;
 $p_{\text{H}_2\text{O}} \cdot s = 15 \text{ mm atm.}$;
 wall temperature $t_w = 300^\circ \text{C}$;

One obtains $\epsilon = 3.8$, and eventually $a_{\text{H}_2\text{O}} = 3.1$ thermal units per m^2/h per $^\circ\text{C}$.

Example : CO₂.

Given : gas temperature $t_g = 1000^\circ \text{C}$;
 partial pressure p_{CO_2} ;
 thickness of layer s ;
 $p_{\text{CO}_2} \cdot s = 30 \text{ mm atm.}$;
 wall temperature $t_w = 300^\circ \text{C}$;

One obtains $\epsilon = 10.4$ and eventually $a_{\text{CO}_2} = 12.5$ thermal units per m^2/h per $^\circ\text{C}$.

N. B. — Gastemperatur = gas temperature. — Wandtemperatur = wall temperature.

where S — absorption capacity of the irradiated surface,

p = partial pressure of the radiating gas in kg/cm^2 ,

s = thickness of the gas layer in m ,

t_g = gas temperature,

t_w = wall temperature.

In order to avoid the mathematical effort, B. KOCH has worked out a nomogram, reproduced by GUMZ [15], p. 79, which can be used for the determination of the coefficients of heat transmission through gas radiation.

9. Summary

In order to simplify and reduce the mathematical effort for the calculation of steam locomotives the principal formulas and data required for the calculation of the locomotive boiler are presented in diagrammatic form. The equation and characteristics used for this purpose are stated, and their choice is justified by comparative calculations.

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The French National Railways order « lengthened » coaches,

by Louis BRUHAT.

(*Chemins de fer*, September-October, 1956.)

The new stock put into service since the end of the war by the S. N. C. F. was built to standard specifications in order to enable the different components of the body to be mass-produced, and therefore manufactured at lower cost. The rather heavy silhouette of the first series, owing to the « petticoat » completely covering over the lower part of the frame was soon lightened by shortening of this « petticoat », and making it a mere « slip ».

From one series to the next, differences crept in, chiefly as regards the interior arrangements which became more polished and elegant, whilst retaining the compartments of classic dimensions, in round figures : 2.120 m (83.46 in.) first class, 1.900 m (74.80 in.) ordinary second class; 1.920 m (75.58 in.) second class sleepers and 1.725 m (67.91 in.) third class.

Until the 1953-1954 stock, all the coaches had the standard length of 23.344 m (76.57 ft.) over buffers.

However, operating with two classes of coaches instead of three was bound to lead directly or indirectly to appreciable modifications to these standards. In fact, as from the 1952 units, in view of the application of the « two classes only » regime, the length of the 2nd class compartments was increased to bring them in line with those of the 1st class compartments in order to facilitate their reconversion at a future date without sacrificing any of the comfort of the 1st class stock. However until the 1953-1954 units, the length of 23.344 m over buffers remained unchanged.

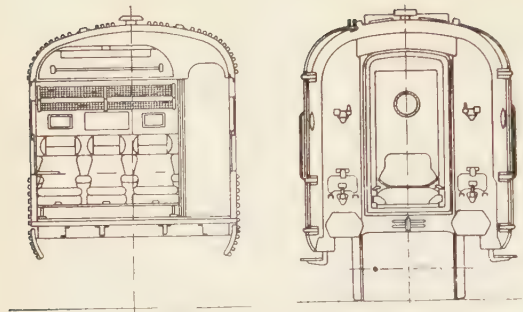
The « two classes only » regime was bound to lead to even greater modifications which, starting with the second class six berth sleeper compartments, had repercussions on all the new stock to be put into service.

In the case of long journeys, night travel has and will continue to have great value as it enables travellers to avoid losing precious day-time hours. But in addition it must be made possible to travel without fatigue. Obviously travelling lying down is the only way to ensure this, and that the formula of lying down berths is the way to ensure this at low cost.

When operating with three classes, the six seater second class sleepers became extremely popular as soon as they were introduced, and time has only confirmed their success as the stock of such coaches has been increased. The « two classes only » regime whilst giving the enormous advantage of allowing people to lie down in the lower class, is bound to increase the extent of this evolution.

If the standard of comfort is to be satisfactory in the case of the second class compartments of today, the length of the compartments must approximate that of the old second class compartments, so that those travelling in the lower class at night can be comfortable. After models had been prepared, the measurement of 1.926 m (75.82 in.) from centre to centre of the cross partitions was finally retained, which is practically the same as the old measurement of 1.940 m (76.37 in.). This

Coupe a



Plan, élévation et coupe des Nouvelles Voitures de 1^{re} Classe A9 allongée à 25,094 m. en acier inoxydable.

d'après documents SNCF

DIAGRAM.

Coupe a = section a.

Plan, elevation and section of the new A⁹ « lengthened » 1st class coaches made of rustless steel. (From S. N. C. F. documents.)

Vue par bout = view from the end.

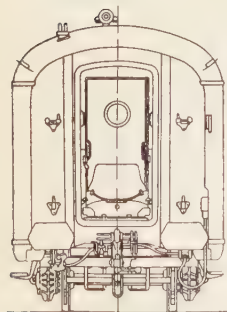
Plan, elevation and sections of new 2nd class B¹⁰C¹⁰ « lengthened » 25,094 m coaches in soft steel. (From S. N. C. F. documents.)

Coupe a = section a. Day arrangement.
Coupe b = section b. Night arrangement.

Vue par bout

Plan, élévation et coupes des Nouvelles Voitures de 2^e Classe B¹⁰C¹⁰ allongée à 25,094 m. en acier doux.

d'après documents SNCF



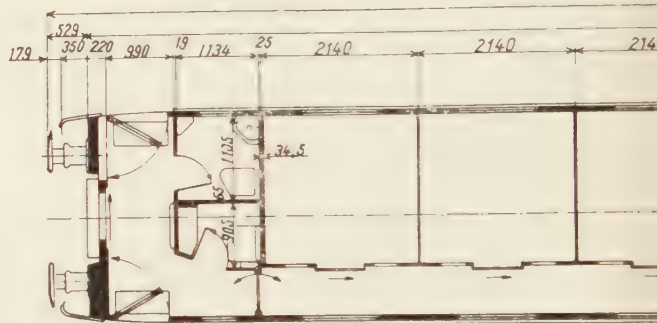
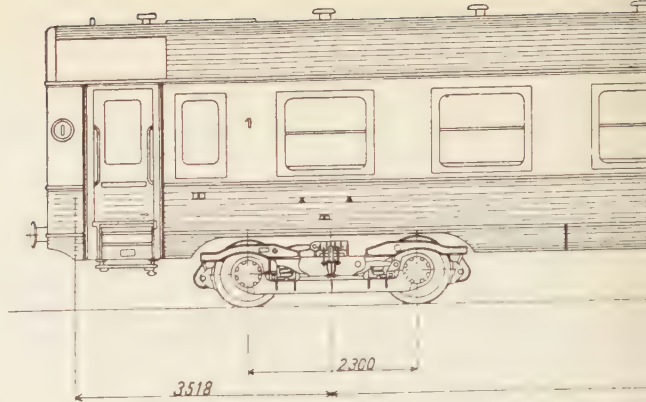
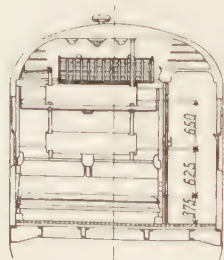
Coupe a

Disposition de jour



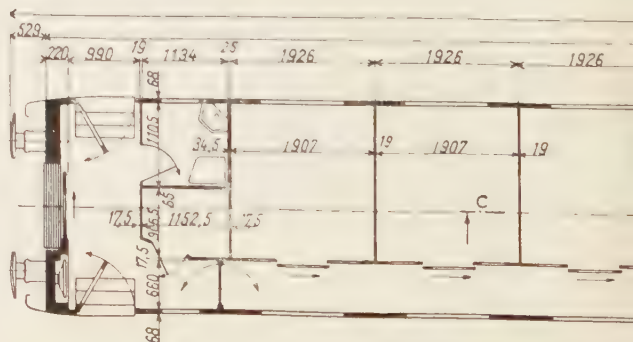
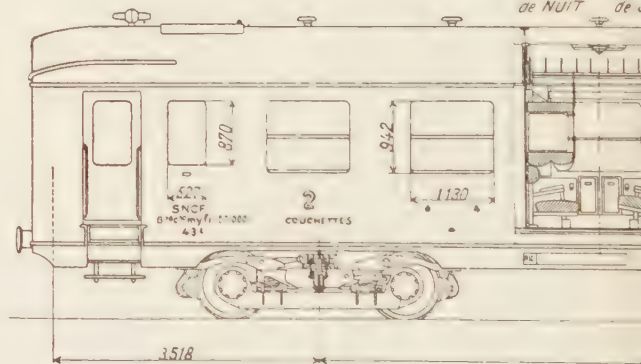
Coupe b

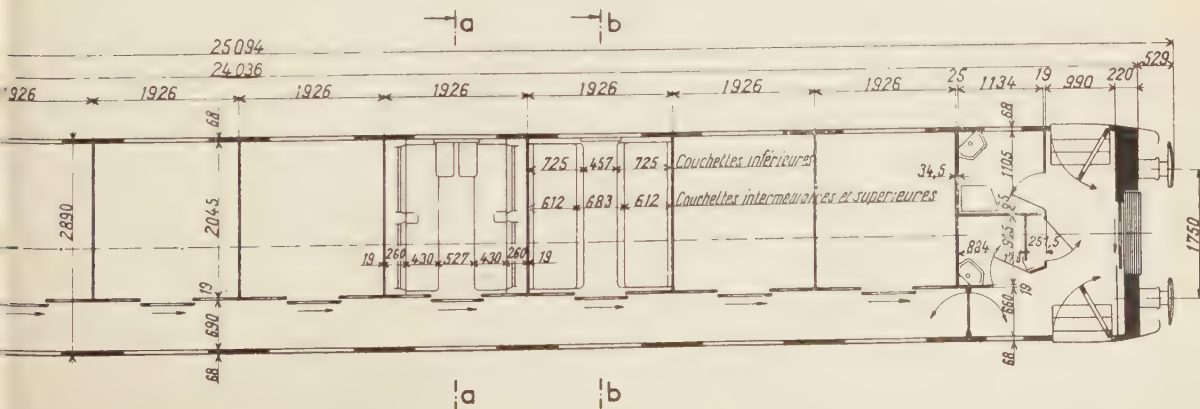
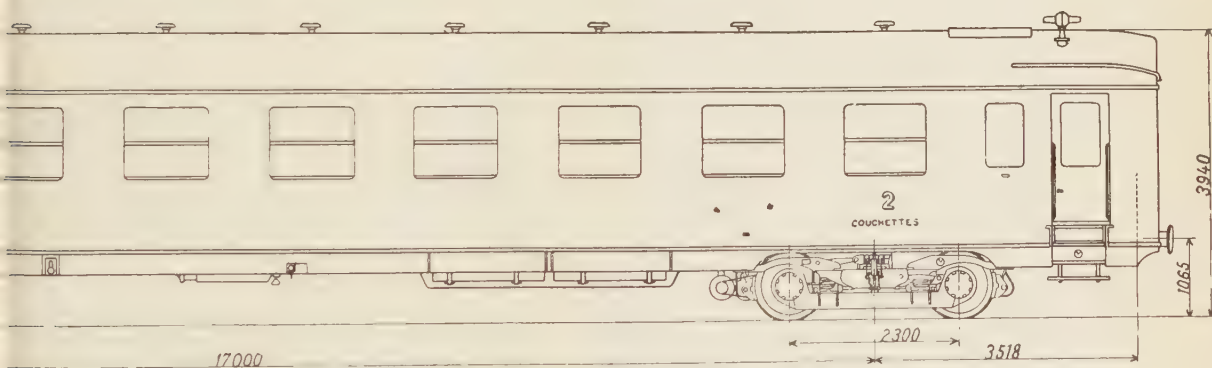
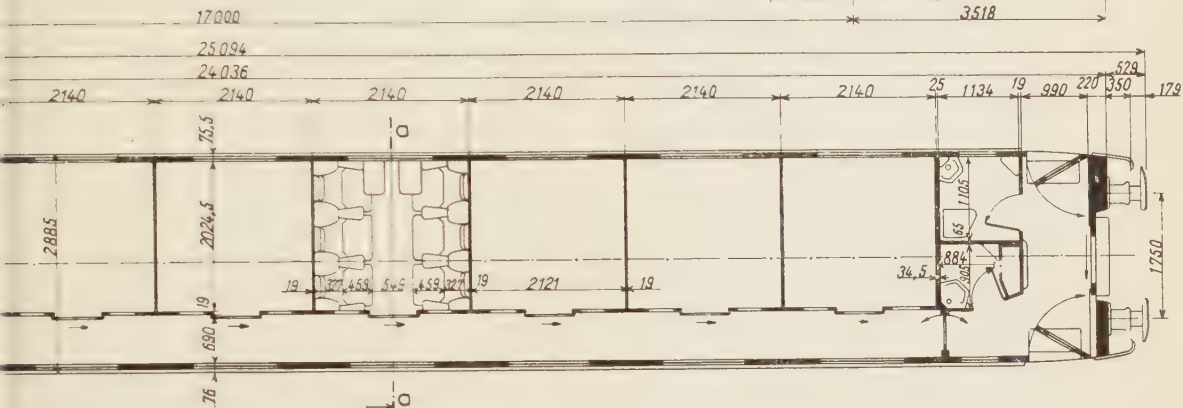
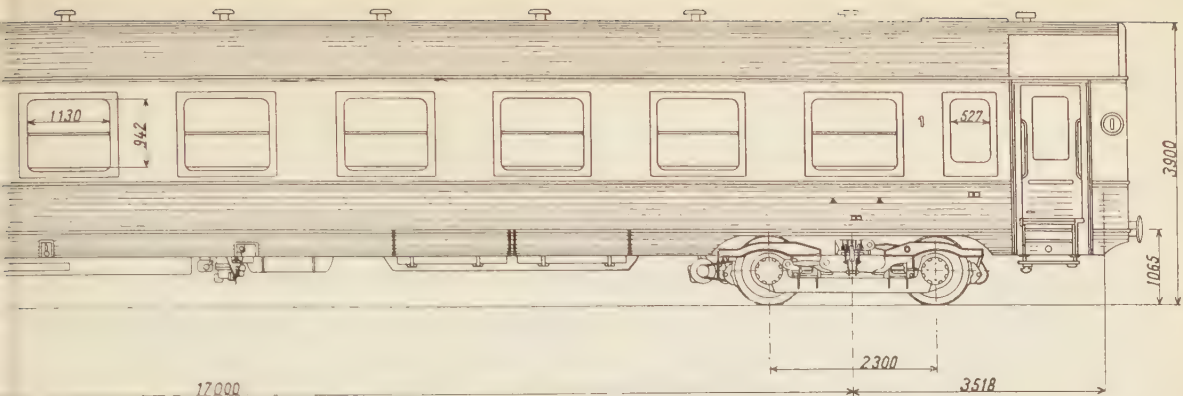
Disposition de nuit



Coupe

Disposition de nuit





increase in the length of the compartments compared with that of the old third class compartments means increasing the length of the coach itself if there are still to be ten compartments. A new coach has therefore been introduced : the « lengthened » coach, 25.094 m (82.31 ft.) over buffers.

Once this length had been adopted for the B¹⁰C¹⁰ coach, it became obvious that for many reasons : standardisation, improved comfort compared with the last C¹⁰ coaches, which were already more spacious, possibility of turning them into « sleepers » at a future date, the same length had to be retained in the case of the B¹⁰ coach.

Apart from the interior arrangements, the B¹⁰ coach is therefore similar in layout to the B¹⁰C¹⁰.

For standardisation reasons, the same length will be used for other coaches, in particular the first class stock which will be known as the A⁹ coach.

The above shows the successive stages which starting from the reason « comfort » have led to a new series of coaches which will now be the standard S. N. C. F. construction : the « lengthened » 25.094 m over buffers coaches.

The diagrams appended show the first coach of the series : B¹⁰C¹⁰ and the A⁹ coach; it should be noted that the new formula applies both to coaches with

bodies of ordinary steel and those of rustless steel made by the Etablissements Carel Fouché.

Owing to inevitable delays in completing the designs, the « 1955 » programme will not be completely of « lengthened » stock.

This programme covers 175 coaches, divided up as follows :

— 23.344 m coaches : 45 B¹⁰ (ordinary steel) and 2 A⁹s (rustless steel);

— « lengthened » coaches : 90 B¹⁰C¹⁰ (ordinary steel) and 38 B¹⁰ (rustless steel).

On the other hand, the 1956 programme, which has just been ordered, includes only « lengthened » coaches :

— in ordinary steel : 135 B¹⁰C¹⁰;

— in rustless steel : 11 A⁹, 7 B¹⁰, 2 A⁹s to get homogeneous equipment in rustless stock for various trains: 8 A⁹ — four special A coaches with kitchens; 4 special A coaches without kitchens; 4 special A⁹s buffet coaches; all destined for the fast Paris-Lille services.

We will deal at a later date with the Paris-Lille stock.

An analysis of the programme underlines the important and continuous efforts made by the S. N. C. F. to ensure the comfort of night travel.

In 1957, the same efforts will be continued, as the S. N. C. F. is contemplating, amongst other things, constructing further B¹⁰C¹⁰ coaches.

The mechanical part of the new single-phase, high-power, motor coach set, series E.T. 30 of the German Federal Railways,

by Eugen LIPPL, Munich.

(*Elektrische Bahnen*, Nos. 9 and 10, September-October 1956.)

1. General.

The construction of main line electric motor coaches always poses difficult questions of coach design.

The electrical equipment, from the pantograph to the traction motor, comprises numerous parts of large volume and considerable weight. It cannot be distributed freely about the body of the coach without affecting running qualities, which are a prime consideration in a passenger carrying vehicle.

However, an electric traction installation which is well proportioned can, even at high powers, be arranged so that the whole of the body above floor level is available for passenger accommodation, almost without restriction, or other service use; it may even be possible to reduce the floor height to eliminate the need for a step or ramp. It is true that, in this case, the electrical equipment located under the floor must be widely distributed, so that its location and erection becomes more complicated and therefore more costly. The cables and wiring must then be arranged in the frame with the utmost care so that it remains readily accessible and does not interfere with the compressed air and oil pipes.

It is not, however, solely the problem of design which makes rail motor coaches the most difficult carriages to build.

The motor coach, from the beginning of rail transport, has been regarded as a more

modern means of transport than the normal train and is measured, even today, in the light of the most advanced technique; it thus demands the construction of a kind of « dream vehicle » for railway work. With each new series, attempts are made to improve on former types, the *operating* qualities in regard to :

- speed;
- acceleration and braking;
- simplicity of service;
- safety of operation;
- multiple operation;
- easy interchange of vehicles and parts;
- accessibility;

running quality :

- smooth vertical and horizontal movement;
- even buffing and drawgear efforts;
- elimination of vibration transmitted from the motor;
- good heat and sound insulation;
- static and dynamic stability of body and running gear;

above all, the special fittings designed to increase *comfort* :

- seats which are comfortable during the journey, scientifically designed, as numerous as possible with liberty of movement;
- technically perfect lighting;
- adequate heating and ventilation with freedom from draughts;
- irreproachable toilet facilities;

adequate space for passengers to pass in compartments, corridors, vestibules and exits:

simple entrance and compartment doors, as far as possible automatic or easy, with safe closure.

The electric railcar, with its very high reserve of power, is particularly subject to these conditions. It is, in fact, the nearest approach to the « motor car of the railway » and is undoubtedly the nearest to providing the passenger with the facilities of a motor car, subject to restriction to a line and timetable, which are much greater than are found in a road vehicle (fig. 1).



Fig. 1. — Front view.

2. New high-power motor coach for single-phase, 16 2/3 cycles 15 kW, series E. T. 30.

In deciding to build a new, high-power, single-phase, motor coach for express inter-city and suburban service, the German Federal Railways have re-entered the field of rail motor coach development at a time when the effects of the war on its own service, and on the German rolling stock industry have been overcome and it is possible to see clearly the technical progress abroad in the construction of self-propelled stock, and taking into account experience also acquired in Germany, have established the principles to follow in the construction of an important class of a new type of vehicle. In addition, work has progressed

with the electrification of lines from the Bavarian system, remote from the mining areas, along important traffic arteries which will end in the great Rhine-Ruhr programme and soon connect the South German system with the North-West.

It was therefore decided to put into service on the whole of the Federal Railways electrified lines — which should, in 1959 (1), reach a total of 3 328.8 km (2 068 miles), a standard type of motor coach which it appeared would have to make up the greater part of the 145 motor coaches required (2). It would have to meet the requirements of workmen's and omnibus trains, heavily loaded, in the Rhine and Ruhr districts, working a semi-underground type of service as well as inter-city and suburban services which are very few in Southern Germany: in addition, it had to be capable of economic use on fast services at speeds up to 120 km/h (74 miles) and, with slight modification, express services up to 160 km/h (100 miles).

For services in industrial regions with a high density of population, the E.T. 30 sets had to meet particularly advanced requirements and indeed, not only improve but revolutionise the quality of service provided on short and medium journeys (3).

These conditions determine the importance and distribution of the sets, the internal equipment and plan of the coaches (fig. 2), the arrangement of seats and doors, as well as the power factors (Table 1).

3. Operating conditions.

The operating programme was based on giving priority to workmen's and suburban traffic in the Ruhr area and suburban traffic in Mainz and Stuttgart. For stopping intervals of :

2.5 km (1.5 miles), commercial speeds of 60 km/h (37 miles), were specified;

(1) KLÜSCHE : « Electric Services of the D. B. in 1955 » *Elektrische Bahnen*, Jan. 1956, No. 1.

(2) KLÜSCHE : « Developments in electric traction on the D. B. after the second World War » *Die Bundesbahn*, May, 1954, No. 9 - 10.

(3) VÖGEL : « Electric traction in modern railway traffic » *ETZ*, edition A. Vol. 73, No. 21.

TABLE 1. — Characteristics of the ET 30.

Length of set over buffers	80 360 mm	
Arrangement of set	Two motor coaches (ET) 'a' and 'b' and one centre coach (EM).	
Axle arrangement	Bo'2' + 2'2' + 2'Bo'.	
Type	BPw 4 ym/AB 4 ym/B 4 ym.	
Draw and buffing gear	Two Sharfenberg centre coupler/buffers at the ends.	
	Two short screw couplers with buffer and plates at the fixed coupler ends.	
Plan	Two driving cabs.	
	One luggage compartment.	
	Seven second-class compartments.	
	Two first-class compartments.	
	Five central vestibules with double doors.	
	Four end vestibules with single doors.	
	Three toilets.	
Wheelbase of driving bogies	3 600 mm	
Wheelbase of carrying bogies	2 500 mm	
	New	Worn
Diameter of driving wheels	1 100 mm	1 020 mm
Diameter of carrying wheels	950 mm	870 mm
Drive :		
Maximum speed	120 km/h	
Average acceleration of max. speed	0.7 m/s ²	
One hour power at 70 % max.	1 760 kW	
Starting power	3 000 kW	
Tractive effort Z_{max}	18 t	
Weight :		
In working order (= empty weight + complete equipment + 150 kg for personnel)	57.5 + 32.3 + 57.2 = 147 t (weight = 146.4 t).	
Payload = Passengers (seated and standing) 75 kg + luggage = ...	34.6 t	
Complete loaded weight	181.6 t	
Weight of body :		
(a) Motor coach		
Without electrical equipment	24.5 t	
With electrical equipment	34.5 t	
(b) Trailer		
Without electrical equipment	19.6 t	
With electrical equipment	20.6 t	
Weight of driving bogies		
Without electrical equipment	9.9 t	
With electrical equipment	17.16 t	
Weight of carrying bogies with equipment		
Motor coach	5.8 t	
Trailer	5.6 t	

TABLE 1. — Characteristics of the ET 30 (continued).

Axle loads :		
Motor coach	Empty	Loaded
Driving axle	18 t	20.5 t
Carrying axle	10.7 t	14 t
Trailer		
Carrying axle	8 t	11 t
Adhesive weight	72 t	82 t
Adhesive weight %	44	46
Weight of electrical equipment :		
Driving bogie		
Motors	5.69 t	
Gears, fans, etc.	1.57 t	
Total	7.26 t	
Body		
Transformer group	6.0 t	
Electrical equipment in body	1.7 t	
Electrical equipment under frame	2.3 t	
Total	10.0 t	
Total per motor coach	17.26 t	
Total per trailer	1.0 t	

4.5 km (2.7 miles), commercial speeds of 76.5 km/h (47.5 miles), were specified;

5 km (3 miles), commercial speeds of 80 km/h (50 miles), were specified,

the average duration of stops being fixed at 30 secs. The maximum speed of 120 km/h (74 miles) was also considered suitable for medium and long distances as well as for fast suburban services. As a precaution, the vehicle portion was designed on the basis of a maximum speed of 160 km/h (100 miles).

This gave the preliminary essential conditions for designing the superstructure and layout. The duration of halts gave the time available for *exchange of passengers*, that is for emptying and filling the coach. The body plans had to provide for a complete change of passengers, assuming all seats and standing places occupied.

4. Access.

If it is assumed that the access vestibules can take in each case a number of boarding and alighting passengers such that the duration of the complete filling or emptying may be considered fixed, the most unfavourably loaded vestibule will, in the case of poor distribution of seats, determine the basis for exchange of passengers.

Extensive research led to the adoption in the ET 30, of a plan in which each access door serves the number of seats and standing places shown in Table 2.

The average boarding load is 30 places served and the maximum, for three doors, is 37. In the case of a complete exchange of passengers, the number to be catered for at one vestibule is 74 persons boarding or alighting in 30 seconds, or 2.5 passengers per second. To determine the possible user of the various types of entrance, note was

taken on a series of types of the times of alighting and boarding per passenger at the hours of peak traffic with a platform 380 mm high (Table 3).

The form and size of the steps and the rise had a major influence on the times of boarding and alighting (fig. 4).

On the basis of a total exchange of passengers only the level entrance, without

steps, of the Hamburg Metro. ET 171 was able to meet the requirements. On the new ET 30 set, however, it was basically impossible to adopt entrances without steps, as on the urban railways, as they would have to be used on all lines and consequently be usable with all platform heights. It was thus necessary to find another solution. Despite the need for space for the electrical equipment under the frame, the height of the floor above rail was fixed at 1150 mm, or 100 mm lower than in passenger carriages. The width of the door passage was fixed at 800 mm, in other words to allow two persons to pass without difficulty and if required, to allow two persons to alight in quick succession. To cater for the difference between floor level and platform, two steps were provided with the normal heights of 260 to 270 mm height and width, to allow freedom of entry and exit even with baggage in both hands. This gave a suitable angle of 45° (fig. 3).

In view of the high acceleration in starting, the doors are closed from the driving position, to ensure passenger safety and to facilitate rapid departure of the railcar sets. By using both faces of the piston, it has thus been possible to use the closing cylinder for electro-magnetic operation of the folding doors. By operating the handle of the outer or inner door the automatic door opener is put into action which rapidly opens the doors to their full width. Because of the large width, the doors fold in four

TABLE 2.

Coach	Vestibules	Number of seats	Number of standing passengers	Total number of passengers
a	1	14	14	28
	2	14	14	28
	3	13	14	27
	4	13	14	27
	5	18	19	37
M	6	18	18	36
	7	17	13	30
	8	17	12	29
	9	16	13	29
b	10	16	13	29
	11	13	14	27
	12	13	13	26
	13	20	17	37
	14	20	17	37

TABLE 3.

	C 3 yg	ET 25	ET 85	C 4 ymg	ET 171
Type of door	hinged	sliding	hinged	hinged	double sliding
Width mm.	880	640	700	600	1 200
Angle of steps (°)	59	48	62	55	no steps
Average time of alighting s	1.3	1.4	1.45	1.4	0.4
Average time for boarding s	1.5	1.4	1.5	1.5	0.4

25 passengers per second, but does allow the average stop of 30 seconds in Ruhr services to be observed with a good reserve, since total exchange of passengers occurs only at

very few stations and the vestibules have been so arranged that they can be used to regularise the movement of passengers in one direction or the other. This assumes that the vestibules are suitable for standing passengers, without draughts, in conditions which compare favourably with compartments as regards noise, heat and safety. The double doors are an essential part of this and the object can be regarded as achieved.

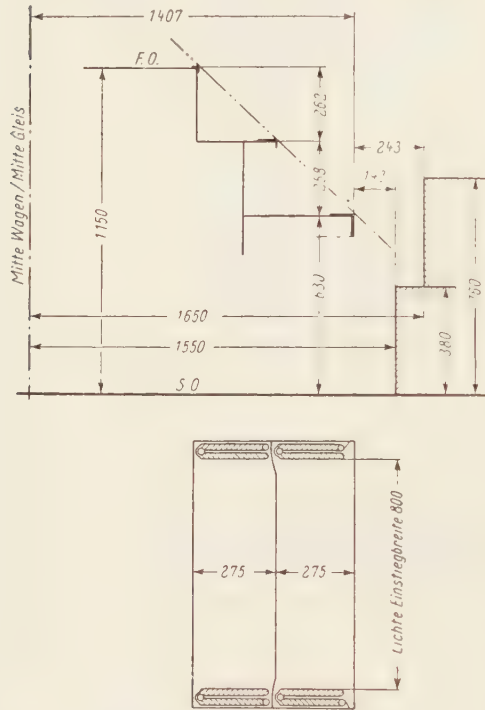


Fig. 3. — Central vestibule

N. B. — Mitte Wagen/Mitte Gleis = Coach track centre, line. — Lichte Einstiegbreite = Free passage width.

5. Protection of gangways.

Protection of gangways should be on the same basis as the end doors. In fact, all the expensive protection already provided in the body could be made useless by poor arrangement of a rigid end coupling.

In spite of all their other advantages, the arrangements already adopted for gangway protection in Germany and other countries are inadequate to prevent draughts, penetration of powdered snow or dust or to give heat and sound insulation whilst allowing through passage with protection against accidents, the latter being essential because of the great increase in passenger traffic and the relatively large movement of carriage ends when rigidly coupled vehicles are passing over points and crossings. The ET 30 set, being used mainly for short distance services, have been given gangways between coaches in the form of an extension of the entrance vestibules. These have therefore been made 1200 mm wide and given four vertical handles. Even on the

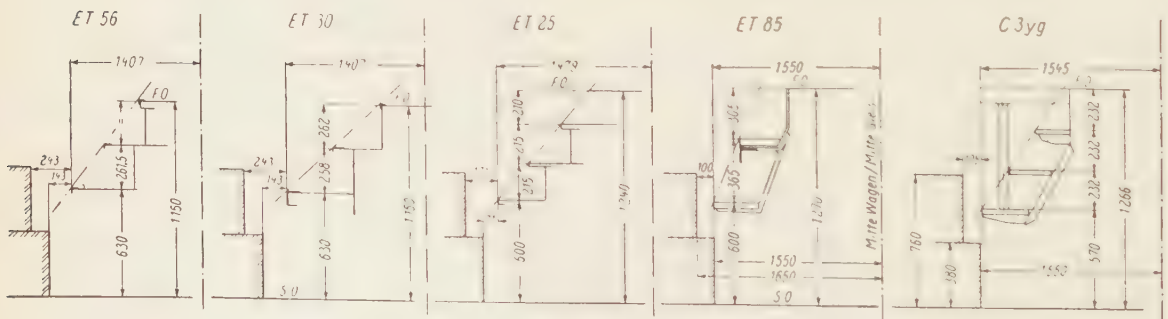


Fig. 4. — Steps for different vestibules.

sharpest curves, the gangways remain covered and lateral movement is restricted by the inner cover.

The layout has been facilitated by the

have the advantages of air-tightness, simplicity and durability together with economy in capital and maintenance costs.

The gangway comprises :

- (1) an internal section with :
 - (a) the gangway itself;
 - (b) the cover.
- (2) an outer cover.

The *inner gangway cover* (fig. 7) comprises a pair of double movable half cylin-

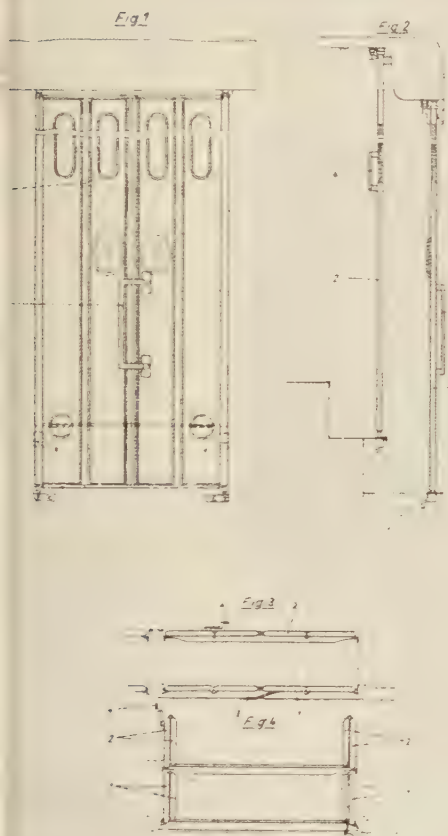


Fig. 5. — Double folding door.

fact that this triplet set has no need for symmetrical division of the gangways. The provision of symmetrical gangways on rigidly coupled carriages, as used on ordinary trains, would have been difficult, useless and inadequate. Moreover, it was possible to fulfil reasonable technical requirements with a non-separable gangway which would



Fig. 6. — Double vestibule.

be made of rubber, like a pair of columns mounted on the end walls of the coach with knuckle-jointed iron fittings to allow them to pivot. The operating handles also serve as handrails for the passengers. The right and left hand parts are interchangeable. With only the inner cover, which itself is as airtight and as simple as earlier arrangements, it would be possible, if necessary, to provide a service equivalent to ordinary trains and allow passengers to pass between coaches as they desired.

The rubber walls can readily be moved from the normal position (fig. 7) to allow the gangway plates to be taken out. In this position there is free access to cable con-

nections, etc. on the end walls. By setting the rubber walls over the door frame the gangway is closed and the coach can be worked separately or without through communication. The rubber walls are sealed

tight shield (fig. 8) and is an extension of the carriage end from the roof to the skirting. It is of corrugated rubber of a new type with small corrugations with ample elasticity without excessive fatigue, and is

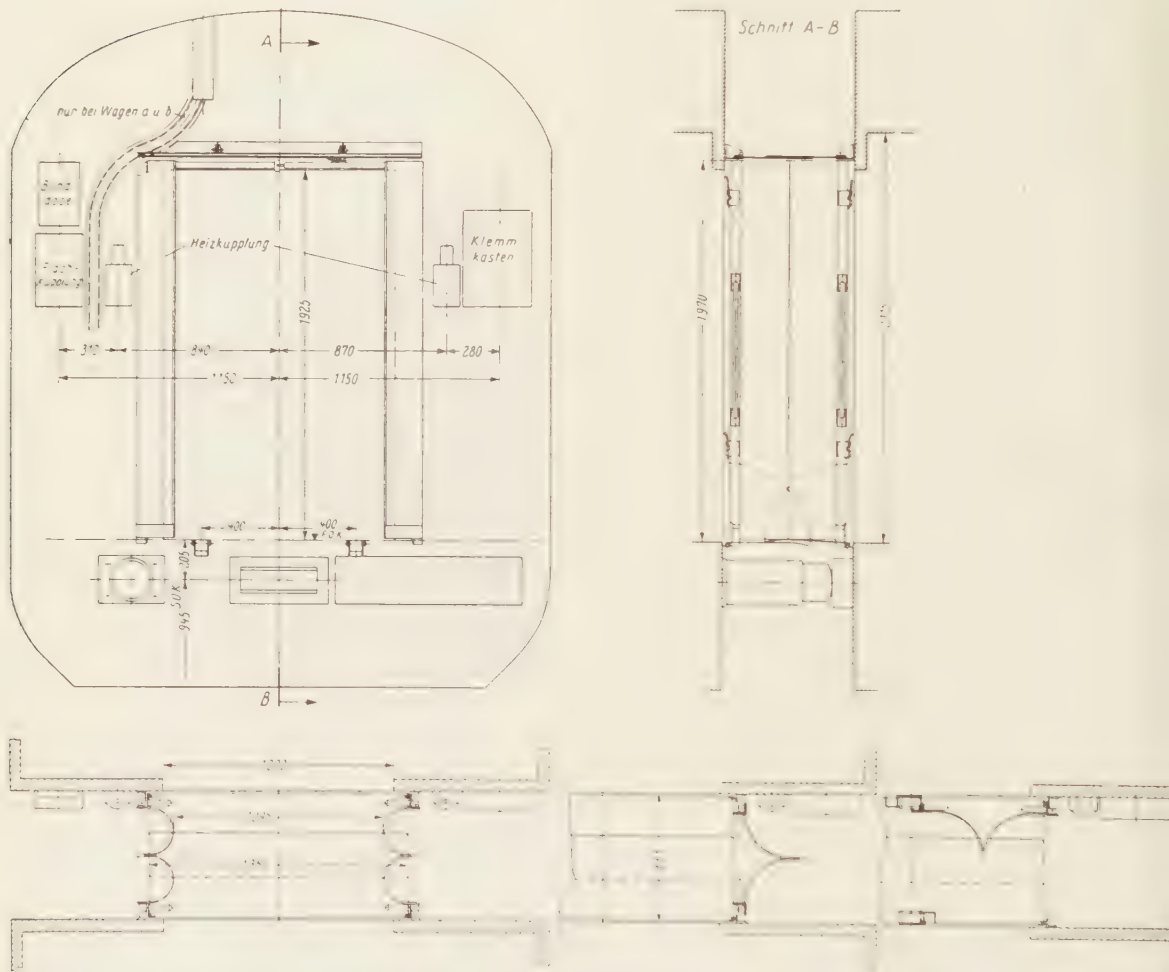


Fig. 7. — Interior gangway protection.

N. B. — Nur bei Wagen a u. b = Coaches « a » and « b » only. — Heizkupplung = heater coupling. — Klemmkasten = Plug box. — Blinddose = dummy coupling. — Flachkupplung = flush coupler.

at the gangway by a soft rubber edging, the top being covered by a drop steel sheet, covered with rubber on both sides.

The outer gangway cover provides a dust-

stable in form. The corrugated rubber fairing is set into a moulded rubber gasket, held between two pressed iron frames. These are fixed to the end walls by gudgeon

pins and keys which would allow quick removal in stations or under the catenary. The protective cover would, of course, then remain suspended from one of the end walls, either the motor coach or the middle coach. A suspension arrangement at roof level facilitates the fitting of the frames. Two side hooks hold the cover in place when vehicles work singly.

and also correct fitting of the whole of the rigid drawgear and outer gangway cover. The greatest relative horizontal movement in the centre lines of the coaches arises on a 190 m (207 yards) radius S bend and increases to 425 mm (16.73 in.) with a drawgear angle of deviation of 28°. These considerable amounts of deviation must correspond also to the rubbing faces of the

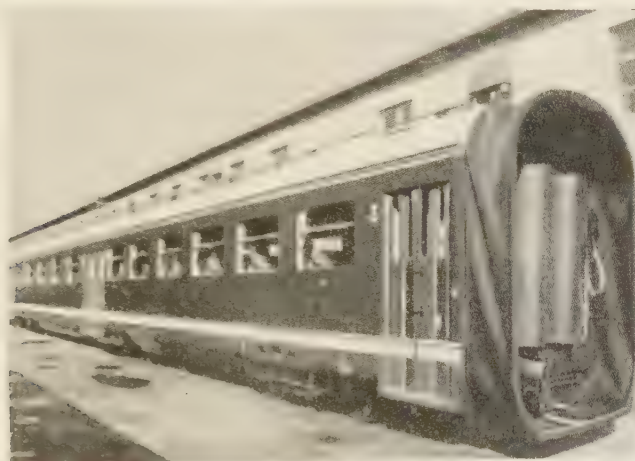


Fig. 8 Exterior gangway protection.

6. Rigid drawgear.

The arrangement of the gangway is closely connected with the *rigid drawgear*. The end vestibules with the consequent blind doors, the necessity for access to equipment mounted on the end walls (cables, compressed air pipes, etc.) (fig. 9), the extension necessary for gangway protection, ease of operation and access to the drawgear itself presented difficult problems in the design of a simple type of drawgear. The type adopted, as being the least liable to accident and simple in operation, is the screw type, able to pivot and bearing on the frame headstock through a traction pivot and cross-bar on the two volute springs. The vertical pivots can be withdrawn from above. This arrangement allows easy release of the screw drawgear

buffers and the buffing plates arranged in pairs on each side. On straight track, the initial tension is 1.9 tons. The separation of the buffers which occurs in very sharp curves up to $R = 125$ m (136.69 yards) has no influence on the running qualities of the set.

This arrangement ensures, in a simple way, easy release to reduce accidents and the ability to tighten the coupling to adjust the gangway. In addition, it damps movement of the vehicle if this is necessary on poor track with worn tyres, although the running gear of the ET 30 does in fact assure automatic stabilisation. In this case, the vehicle bodies are stressed only in the parts designed to absorb draw and buffing forces. Line tests have shown that apart from the relative movement geometrically caused by passing over curves or track

equipment there is no transverse or rotary movement which would affect passengers standing in the gangway. The latter can, as envisaged, therefore be used by passengers boarding or alighting thus providing identical capacity for the two end vestibules.

7. Passenger compartments.

After the vestibules, it was necessary to arrange for the *passenger compartments* to have as equal a distribution as possible.

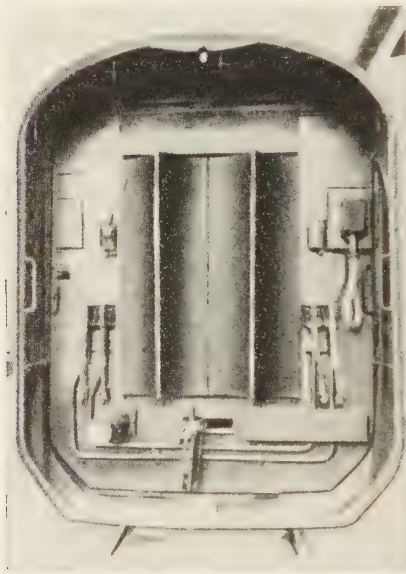


Fig. 9. — Fixed drawgear.

Complete freedom of distribution is limited by the fact that the vestibules are controlled by :

- the location of the driving and carrying bogies having wheelbases of 3 600 and 2 500 mm respectively;

- a type of bogie frame (affected by the use of shoe or disc brakes and driven by frame or nose suspended motors;

- the pivot spacing of 19 000 mm.

Equally important was the installation of the main parts of the electrical equipment

(transformer, fans, ducts, equipment boxes), the driving cabs, and baggage compartments. The provision of a separate postal compartment was not proceeded with. On the twin sets one of the baggage compartments can be used as a postal compartment. No special compartment is provided for passengers with luggage, particularly as the baggage compartment which is fitted with folding seats, can be used for this purpose. The first class accommodation in the middle vehicle has also some influence on the arrangement.

The fact that the ET 30 will be used mainly for services where there is frequent change of passengers called for the greatest possible uniformity in the distribution of compartments in relation to the vestibules and the use of large compartments. This development now seems to be more and more appreciated, despite some initial delay, by the main European services. The adoption of a central corridor was thus inevitable.

The discharging time imposed and the capacity of the set governed the number of exits (14 on each side). The double exits having a width of 2 110 mm and the end exit 1 263 mm, they occupy in total :

$$5 \times 2\,110 + 4 \times 1\,263 = 15\,602 \text{ mm,}$$

of the total length of the train.

The two driving cabs ($2 \times 2\,265$ mm) and the baggage compartment (3 070 mm) in turn take up 7 600 mm not available for passengers, therefore, are :

$$15\,602 + 7\,600 = 23\,202 \text{ mm,}$$

of the total length of the train.

With the body lengths used, there remains for the passenger compartments the space shown in Table 4.

In the 22 m coach (pivot spacing 16 000 mm), there would be room for three first class passenger compartments of about 2 100 mm or 6 300 mm and in the 26 m coach (pivot spacing 19 000 mm), there was a total length of 10 500 mm for the desirable five first class compartments, especially for Ruhr services. There remained, therefore, for the second class, in round figures 39 and 45 m respectively.

TABLE 4.

Pivot spacing	Body lengths	Total length of set	Driving cabs and luggage compartments	Length of passenger compartments	
				mm	%
16 000	2. 22 940 - 22 600	68 480	23 202	45 278	67
19 000	2. 26 375 - 25 760	78 510	23 202	55 308	71

The distance between facing seats of 2 100 mm (first class) and 1 700 mm (second class) are about right for present day standards of comfort. It was thus possible, even in the second class to use the anatomically correct seat recommended by AKERBLOM. In this class, the upholstered backs of the seats, which has become the general rule, call for a longer compartment than the former wooden seat (1 500 to 1 600 mm) to allow sufficient knee-room.

In the set of 22 m coaches, it is therefore possible to provide twenty-three second-class compartments and in the 26 m coach set to provide twenty-seven compartments.

The number of passengers per compartment is governed by the width of the vehicle and by the width of seat considered desirable. The permissible widths for the two types of coach body were fixed at 2 925 and 2 814 mm respectively and their internal widths at 2 805 and 2 694 mm. In the modern large-compartment D.B. coaches, the basic sizes used for seat widths in the second class are about 470 to 520 mm. It must be remembered that the seat sizes are measured between arm rests along the front or rear edge of the cushion and possibly with a certain amount of space from the side wall.

The widths of the corridors are measured between arm rests, seat cushions or luggage rack carriers. The dimensions vary between 480 mm and 730 mm. In the two types of coach in question a corridor width of 500

mm would leave 2 305 and 2 194 mm respectively for seat width.

In the 22 m coach, a 3 + 2 and 2 + 2 arrangement would give seat widths of 460 and 575 mm respectively. The first has been used many times, mainly with arm rests and without the standard of comfort expected in a modern high-power motor coach. With 23 ten-seater bays there would be 230 second class seats. The second case would give a larger seat suitable for first class. It would reduce the number of second class seats to 184.

The 26 m coach, which only provides for a 2 + 2 arrangement gives a comfortable seat width of 550 mm. In the 27 eight-seater bays there is space for 216 passengers.

Comparison of the capacities (Table 5) shows clearly the advantage of the long

TABLE 5.

	Distribution of seats	Number of seats		
		2nd class	1st class	total
22 m coach	3 + 2	230	18	248
	2 + 2	184	18	202
26 m coach	2 + 2	216	30	246

body in regard to possible number of seats at the present-day standards. It is, of course, necessary to deduct toilet accommodation from these figures, but the general table is not changed. It may, however, be mentioned that the minimum changes in the number of vestibules and seat arrangement would be enough to give a completely different picture, so that the above remarks on layout would have to be re-stated in each particular case.

In the case of the ET 30, the above reasons justify the adoption of the 26.4 m body. It has advantages because of the high number of uniformly spaced vestibules.

8. Seats.

Once the plan was settled, the interior fitting could be examined in detail. Severe weight limitation called for lighter construction than any yet used, which led to the adoption of a new type of seat for larger compartments, with transverse luggage racks.

The *second class seats* are mounted on a tubular steel frame with thin partitions, combined in the double seats with baggage and umbrella racks, independent on the corridor side and fixed directly to the side wall. The single baggage rack pillar is designed as a rigid stay. The arm rest assemblies are used to increase rigidity. It has thus been possible to avoid the pillars extending to roof level, which improves the internal appearance. The shape of the seats has received careful consideration. These must be fitted up with all the more care as the arrangement of the compartments has come disadvantages compared with the type of seats provided in other countries. It is impossible to apply a general standard to seat comfort, because in railway practice it is difficult to use individual seats which are adjustable in depth, width and position. It must also be noted that in the long-run the anatomically correct seated position does not always at first appear the most comfortable. Moreover, correct seating is judged by non-medical people subjectively, without regard

for sufficiently general orthopaedic principles, applicable to a large number of people over a period of some hours.

Based on the well-known seating profile established by Dr. Bengt AKERBLÖM and a bay width of 1 700 mm, a form of construction has been designed to avoid, as far as possible, fatigue whilst largely satisfying the many conditions imposed by the vehicle (fig. 10).

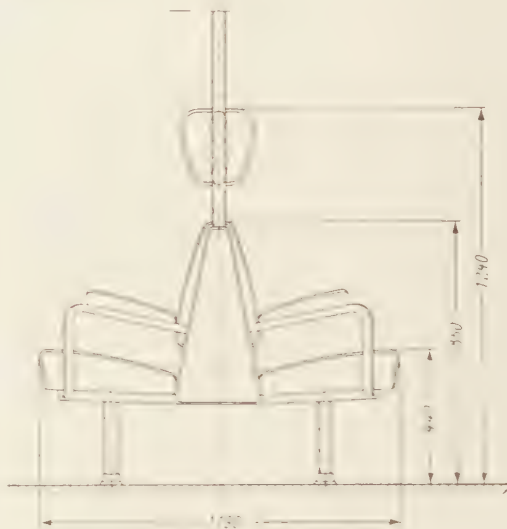


Fig. 10. Double second class seat

The depth of the seat is 595 mm to the back wall and 470 mm to the back rest, the distance between cushions is 510 mm and the height of the seat at the front edge of the cushion is 440 mm. The width between arm rests, for two seats, is 995 mm at the front, and 938 mm at the back, 1 065 mm from the edge of the cushion to the wall and 1 105 mm from the arm rest to the wall, or 532.5 and 552.5 mm per seat respectively. These dimensions are the largest used at the present time for comparable types of seat. Experience in service has been favourable to the opinion that they are generally satisfactory. From the dimensions of the seats, the corridors are 484 mm wide, between the arm rests and 564 mm

between cushions. This provides ample room for passengers to move about or stand in the corridors and makes heavy loading more bearable (fig. 11).

This result has only been obtained by using a most up-to-date design of frame and trimming. The cushions are carried on a



Fig. 11. — Second-class compartment.

tubular steel frame with wire mesh and wooden mouldings. The 40 mm foam rubber padding is covered with imitation red leather of a morocco finish. With its numerous shallow cavities, this design has given good results, not only as regards surface appearance, but also as regards marking and deterioration, where tests have shown it to be very good, particularly by comparison with heavily embossed types.

Also trimmed with the same materials are the back rest, mounted on a light wooden frame with flat spiral binding and cushioning of rubber cord, the head rest of round foam rubber, 15 mm thick, on a trimming board with hollow frame down to the back rest and veneered ply between the back rest and the seat. In addition, to a pleasing

appearance and as high a standard of comfort as is compatible with the light weight construction, they had to have an economic cost. With a weight reduction of about 25 % and a cost reduction of about 30 % on comparable modern types, they have given the required results.

In the *first class* a type of lateral head rest has been designed for large compartments and used so far on Diesel railcars. These seats are very suitable for the compartments. The head-rests are vertical, the seat adjustable horizontally. The latter are mounted on tubular steel carrier frames of light section and wood frames with woven springs. The padding comprises a layer of springs covered with foam rubber on webbing and covered with moquette. The luggage rack is of square tubes of light anodised metal brass coloured. On the corridor side is wood covering the frame, but this is not used on the wall side. The central arm rest is fixed and the backboard is continuous.

With a compartment width of 2 126 mm the distance between the seat backs is increased to 631 mm, the depth of the seat backs to 740 mm, the maximum width between arm rests to 535 mm and the width of the seat back, without a gap between the wall to 625 mm. The width of the corridor

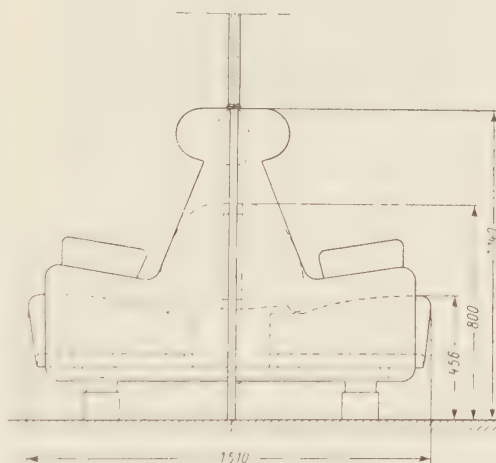


Fig. 12. — Double first-class seat.

has been conveniently maintained at 619 mm between arm rests, 699 mm between the end boards of the seats and 774 mm between the luggage rack pillars (fig. 12).

9. Interior decoration.

For the remainder of the *interior fitting* an attempt has been made, in the *second class*, for example, to give a bright appearance to the compartments and to avoid, by a lighter and less substantial appearance, the conservation which has become almost

for the second class season-ticket holder who used it daily, both in daylight and under fluorescent lights. At the same time, it was necessary to avoid the risk of stains and provide for easy cleaning. In the second class, the floor is covered by brown linoleum and the lower part of the walls by marbled linoleum. The inner doors and wood mouldings are of light oak. The partition walls are covered with sand coloured imitation leather with a surface grain. The top stays are of aluminium-coloured light metal and lie against the ivory coloured ceiling.

In the *first class*, discreet efforts have been made to provide as pleasing and agreeable a compartment as possible, having little of the traditional railway style and giving a cleaner and more restful appearance. Light and highly polished sides of African pear-wood with German pear-wood mouldings assist in creating this impression. The ceiling, separated from the walls by the brass coloured light metal beading, is of polished natural beech veneer. The floor is covered with grey linoleum. Small fixed tables are provided on the side walls. A hinged door, amply glazed, separates the smoking and non-smoking first class compartments (fig. 13).



Fig. 13. — First-class compartment

standard practice in railway carriage construction as regards style and colour, without appearing too « contemporary ».

It was necessary to start with some given colours, for example those of the floor and walls. In styling, it was necessary to avoid anything which, according to present-day ideas could also at the same time have appeared old fashioned. The clear and comfortable appearance of the compartment interior had to be maintained, especially

The *sliding compartment doors* have automatic closing. The closing device is a light lever, of «U» profile, jointed on a pivot in the door housing. The lever bears on the rear edge of the door, by a roller, through springs and returns the door by a slight, adjustable, pressure to the closed position. The door, when fully opened, will stop in the open position. A slight pressure releases it and it will then return to the closed position (fig. 14).

The *windows*, which are of a type used in modern railcars, are divided horizontally and open upwards by a winder handle; they have a uniform width of 1 200 mm. The decision, vital to the bodywork, to use in the second class the window sizes so far used only for first class, was prompted by the desire to raise the standard of comfort generally in the motor coaches. The same desire led to the provision of large windows

in the cab cross partition of coach *b*, to give a view of the track through the driving cab and make the leading compartment attractive by giving passengers a view of the driving operations.

The luggage compartment of coach « *a* » has a floor area of 5.5 m². Above the motor fan ducts and tool boxes are four reserve seats. The luggage compartment has on each side, two leaved hinged doors, 1075 mm wide, which open inwards to avoid risk of accident on the platform. In coach « *b* » it was possible to locate the ventilation ducts under the seats.

Each coach has a toilet with the usual equipment, with entrance from the vestibule.

10. Lighting, heating ventilation.

Lighting has been given particular attention. The choice of fluorescent lights was governed by present day lighting technique. The problem was to arrange the lighting fittings harmoniously and as discreetly as possible in the passenger compartments, vestibules and other areas, to give an architecturally satisfying effect and provide uniform intensity and distribution without harshness. At the same time, it was necessary to ensure easy fitting and access and a judicious gradation between compartments, vestibules and platform lighting. The radiating surfaces of the lighting fittings had to be uniformly illuminated without causing objectionable shadows, such as might be caused by the lampholders. Use was made for the first time, of two longitudinal rows of lamps, dropped 75 mm below the central part of the ceiling.

The light fittings take square reflectors, the vertical surfaces of the plexiglass directed towards the arched part of the ceiling whilst the horizontal parts direct light downwards. The curve of the roof acts as a reflector so that the passenger has an agreeable combination of direct and indirect light at a high intensity. The good appearance of the compartments is thus maintained under artificial light, in spite of the powerful illumination of 130 lux in the second class and 140 lux in the first class,

and its uniformity — the variations being less than $\pm 10\%$. Thanks to the drop in the centre of the ceiling, there is also a gradation of the roof lighting which avoids the monotony of a large illuminated area, without causing objectionable shadows. The optical effect of the illuminated roof curves gives the impression of an enlarged compartment. In addition, there is an effect, although there is a large compartment, that the bays are separated from the central corridor, which must be the result of the lighting. Harmonising the colours, particularly of the seats, with the lighting

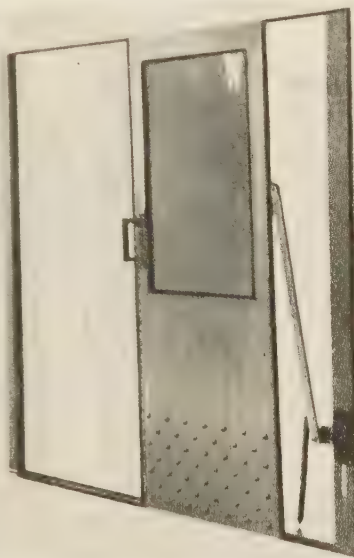


Fig. 14 — Sliding compartment door with spring closer.

and choosing the shades with care has given the compartments a standard of lighting which is very near to natural light.

To the right of each vestibule is a lamp enclosed in the ceiling which also illuminates the steps and, as it is directed crosswise, does not dazzle the passenger entering.

The two compartment lights, 40 W in the first class and 25 W in the second class are controlled by a common switch. Each of

them has an emergency 5 W filament lamp. The toilet compartments have 25 W lamps. The side reflectors of the passenger compartment fittings fold back to the ceiling, as do the intermediate units which link them together. In the latter, readily accessible, are the switches, capacitors, emergency lighting transformer and connections.

The lamps are supplied at 200 V A.C. by a converter (1 500 VA) connected to a 110 V battery.

11. Driving cabs.

The *driving cabs* (fig. 15) have been designed with special care. The arrangement of switches in cabinets at window sill level and in the panels at the right and left gives a very clear layout, so that here too it has been possible to provide very pleasant and neat accommodation for the driver and mate.

To provide for possible future working



Fig. 15. — Driving cab.

Electric heating for the passenger compartments is provided by tubular radiators under a metal grill along the side walls. In the vestibules, luggage compartments, toilets and driving cabs — here in the two sides of the control panels and in the two side walls — are bar heaters in flush boxes. The consumption is 25.5 kW (26 in coach « b ») and 28.5 kW in the driving coach.

Ventilation is provided by static fittings operative in either direction, uniformly distributed, comprising one or two air ducts in the lowered portion of the ceiling with grid inlets. The driving cabs are also amply ventilated.

in multiple and to allow staff, at least, to pass, end doors would be needed. They have not been fitted so far, but they could be provided without major alteration. The driving cab has seven high windows; the end two can slide downwards.

In each of the two cabs there is a door to allow the driver to pass into the luggage compartment (coach « a ») or the passenger compartment (coach « b ») in addition, in coach « b », there is a special vestibule for guard's use when the train is crowded. In coach « a », the luggage compartment door can be used for this purpose.

(To be continued.)

Examining the curve running of railway vehicles by strain measurements in the track,

by F. DIETZE, Dipl.-Eng. Magdeburg.

(*Deutsche Eisenbahntechnik*, March 1956.)

1. Introduction.

In current practice, the running of railway vehicles through curves is treated purely mathematically and graphically as a geometrical problem, using the methods of ROY, VOGEL (1) and SCHACHUNJANZ (2). With these methods, the transverse play of the axles and the gauge amplification are taken into account, but not such as other factors as the dynamic influences of the vehicle during the run, the elastic or permanent strains of the rails and axle springing, etc.

The present notes deal with a method which permits the examination of the behaviour of a railway vehicle in a curve by determining the strains imposed on the rails by the vehicle, or rather the stress distribution over the cross-section of the rails. The strains are measured. The vehicle may run through the curve at any desired speed so that one is also able to ascertain, and follow in detail, the interaction between the dynamic forces of the vehicle and the elastic deformations of the axle springing and the rails.

In the following, the method will be explained with the aid of some examples; in a later publication, we shall apply it systematically to a given type of vehicle and shall, inter alia, provide quantitative data on the forces exerted by the vehicle on the track.

2. Strain gauging method.

In order to investigate the running of a vehicle through a curve with constant radius, constant gauge and constant super-

elevation, it is sufficient to know the strain distribution in one cross-section of each rail (outer rail and inner rail). These two cross-sections must be situated in the same plane; in other words, they must be exactly opposite each other. Where curve radius, gauge and cant vary, as well as on points and cross-overs, the strain distribution must be measured in several planes. Number and position of these planes depend on the extent of the variation of the curve characteristics along the curve.

In the examples here used to illustrate the practical application of the method, it has been possible to assume with sufficient approximation that curve radius, gauge and cant were constant in the vicinity of the gauging section, viz. over a length of curved track corresponding to at least one vehicle length in front of the gauging point, and one vehicle length behind it. In the circumstances, it has been possible to investigate the behaviour of the vehicles in the curve merely by measuring the strain distribution in a *single* plane. This plane was in the centre of the curve, equidistant from the nearest sleepers. For purposes of comparison, the strain was also measured in each rail at a point outside the gauging plane proper, viz. at a point half-way between the next and the next-but-one sleeper.

The strain distribution in the rail cross-section is determined from the elongations measured on the rail surface, i.e. at the edge of the cross-sections. Here, as well as in the remainder of these notes, the term « elongation » is also meant to cover

compressive strains (« negative elongations »). To calculate the stresses from the elongations, a modulus of elasticity $E = 2.1 \times 10^4 \text{ kg/mm}^2$ has been assumed. In conformity with Hooke's law, the distribution of the stresses over the cross-section is assumed to be linear. The values found at the surface are thus connected by straight lines.

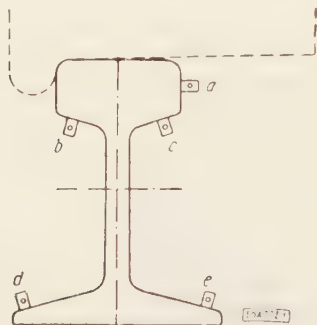


Fig. 1. — Strain gauges distributed over the measured cross-section; *a*, *b*, *c*, *d* and *e* are measuring points.

The stresses, or rather the elongations, on the surface were determined at five points of the gauging section (cf. fig. 1) viz. on the outer side of the rail head (*a*), on either side below the rail head (*b* and *c*), and on either side on the rail flange (*d* and *e*).

Figure 2 shows the three detectors on the outside of the rail, with their connecting lines.

The elongations were measured with the aid of strain gauges of 10 mm length, designed by Dr. LANGE (cf. fig. 3). The detectors rest on small supports which are pasted to the piece to be measured, i.e. in this case, to the rail. As an adhesive suitable to fasten steel to steel, « Holl-fast » has been found to give very satisfactory results. In the case of an elongation, a transverse armature moves in the detector between two coils, causing a variation of the detector inductance and, in consequence, the detuning of a Maxwell bridge which had previously been tuned

in the zero position of the detector. The electric current produced by this detuning is amplified by means of a measuring frequency modulator with a carrier frequency of 5 000 cycles, and recorded by a loop oscillograph. The installation is calibrated before and after the measurements.

As regards the measuring technique adopted, it must be pointed out that, owing to the use of amplifiers, it is still necessary to reckon with faults, and the measurements here discussed were also affected by such faults. This is why we do not, in every case, state the results of the measurements taken at all five gauging points of the cross-section.

3. Influence of a transverse force on the stress distribution in the rail.

The behaviour of a vehicle in the curve is characterized by the position which the vehicle assumes in the curve, by the wheels running up against the outer rail and the inner rail respectively, and by the way in which the vehicle acts on the rails. This action varies according to whether

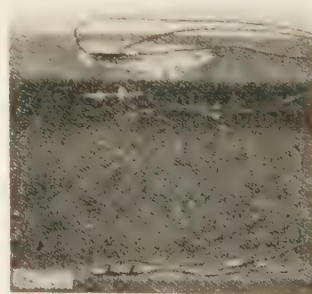


Fig. 2 — Strain gauges with connections on the outside of the rail.

the wheel merely rolls on the rail, or whether there is also a transverse action of the wheel flange on the rail. In the former case, the wheel pressure is mainly exerted perpendicular to the rail tread. However, as there is always a measure of

friction between rail and wheel, the axle pressure is never exactly perpendicular to the rail tread. If the flange is also engaged, the guiding force acting on the side of the rail head must be added to the axle pressure so that the resultant of axle pressure and guiding force is inclined in relation to the plane of symmetry of the rail. In that case, the rail is subjected to bending stresses, torsional stresses and shear stresses.

subjected to a bending moment M_b can be calculated from the equation (1) :

$$\sigma(x, y) = M_b \cdot I_y \cdot \lambda(x, y),$$

where

$$\lambda(x, y) = y \cdot \frac{\cos \alpha}{I_x I_y} + x \sin \alpha.$$

As we are here merely interested in the distribution of the stress, $\lambda(x, y)$, and not in its value $\sigma(x, y)$, no scale has been



Fig. 3. — Strain gauge, with and without stop pin.

Because of the very great difference in the bending resistance of a railway rail in relation to the two axes of inertia of the cross-section, the stress distribution is very sensitive to any obliqueness of the load so that, inversely, it is easy to deduce from the stress distribution whether the load had acted essentially in the direction perpendicular to the rail tread, or if a more or less pronounced transverse force had taken part. To illustrate this condition, figure 4 shows the calculated distribution of the normal stresses for different angles of the resultant load. In this example, the rail is subjected to bending stresses by a concentrated load forming an angle α with the axis of symmetry of the cross-section. For simplicity's sake, it has here been assumed that the line of application of the load passes through the centre of gravity of the cross-section. Normally, this is not the case, so that the rail is also subjected to torsional stresses. For the present, purely qualitative considerations, this simplification may be permitted.

In the case of pure bending stresses, the stress distribution in a cross-section

chosen. The influence of the ratio $I_x I_y$ of the main moments of inertia of the cross-section can be recognized. The greater this ratio, the greater is the influence of the second term of the distribution equation which represents the influence of the transverse force. As will be seen, it is not necessary for the load to have a considerable obliqueness before, for instance, tensile stresses appear on the outside of the rail head, and compressive stresses on the inside of the flange. Normally, i.e. with vertical load, one merely expects compressive stresses in the rail head and tensile stresses in the rail flange.

In practice, such stress distributions and stress variations can occur in rapid alternation as will be seen from the recordings reproduced in figure 5. These stress distributions have been recorded in the curve of a metre-gauge track. The radius of the curve was 60 m, the gauge amplification in the immediate vicinity of the measuring point 20 mm, otherwise 15 mm,

(1) Cf. for ex. HÜTTE, vol. I, 25th edition, p. 589.

and the superelevation 20 mm. The track was equipped with Type 5 rails on Type H permanent way. The stress distribution was produced by a narrow-gauge B + B locomotive passing the measuring point in the forward running direction at a speed of about 25 to 30 km/h.

The oscillograph recordings represent what may be termed « stress influence lines » (or, more accurately, « multiple stress influence lines »), i.e. each of the six curves always records the stress at the same gauging point while the different axles of the locomotive approach this

the peak stresses are proportional to the distances between the axles.

The maximum values of the upper curve (I/1) always show a constant lead in relation to the maximum values of the other curves because this curve corresponds to a measuring point on the side of the rail head located ahead of the measuring plane proper halfway between the two preceding sleepers.

If we first confine our considerations to the upper chart which indicates the stresses at three points of the outer rail, it will be seen that, on the outer side of

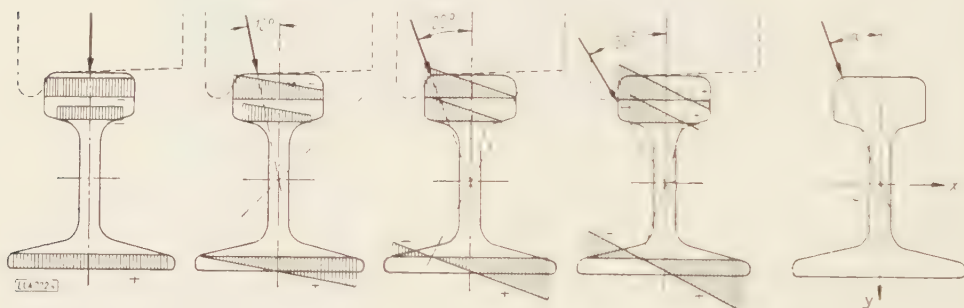


Fig. 4. — (Calculated) stress distribution in the rail cross-section under an inclined load.

point, pass over it (maximum values), and move away from it again. As the paper of the oscillograph moves on, at a uniform speed, the recording is spread out on a time scale at it were. If the speed of the locomotive and that of the paper feed is known, it is possible to ascertain, after the event, the position of the locomotive for any desired stress distribution.

The heavy peak stresses, especially those measured on the rail head, always occur when an axle is directly on top of the measuring point. Because of this fact, it is possible to correlate the position of the load with the stress amplitude so that each peak stress can be marked with the number of the axle which has caused this stress at the measuring point. If both the paper feed and the speed of the locomotive is constant, the intervals between

the rail head (I 1) and on its underside (I 3), the first axle of the locomotive causes tensile stresses whilst the following axles cause compressive stresses. On the inside of the rail flange (I 2), one finds, under the same conditions, first compressive stresses and then tensile stresses.

The stress distributions under the first axle of the locomotive are very similar to those distributions shown in figure 4 where the resultant has its maximum inclination, the only difference being that the measured stresses are even more pronounced. In reality, the angle of inclination and, consequently, the transverse force must therefore be even greater. From the stress distribution caused by the first axle of the locomotive, it can be concluded with certainty that there must be a transverse force which can only be caused by

the wheel fouling the outer rail. It is impossible for a side force of this magnitude to be caused by the transverse friction between wheel tread and rail alone. Moreover, the transverse force resulting from friction on the outer rail is directed inwards, i.e. in the direction opposite to that of the guiding force.

In the case of the stress distribution caused by the other axles, the axle pressure is seen to act mainly perpendicular to the rail tread so that there are no

compared with that caused by the two following axles. Here, the transverse force is obviously caused by the transverse friction which, on the inner rail, is directed outwards. That a fouling of the inner rail by the first axle is impossible can already be concluded from the fact that this axle has already been found to foul the outer rail.

The transverse forces exerted on the inner rail by the two following axles are somewhat greater. This is also borne out

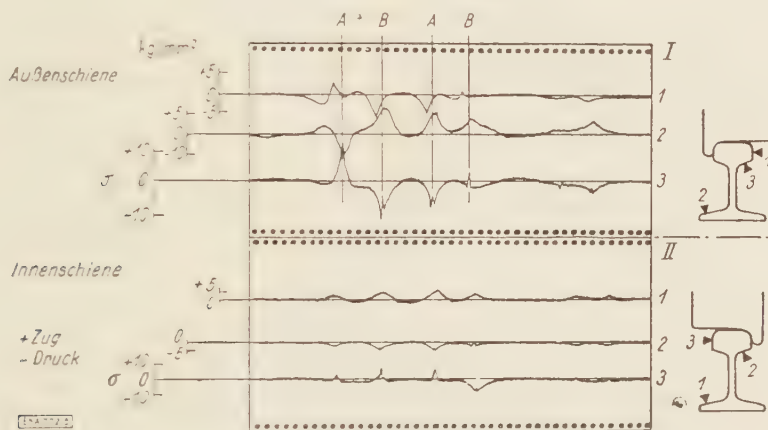


Fig. 5. — Stress distribution under a narrow-gauge locomotive B + B in a curve of 60 metres radius. The different curves correspond to the measuring points bearing the same code number.

N. B. — Außenschiene = outer rail. — Innenschiene = inner rail. — Zug = tensile stresses. — Druck = compressive stresses.

transverse forces caused by the wheel flange. These axles therefore do not foul the outer rail.

Similar considerations reveal the position at the inner rail. The lower chart of figure 5 shows three stress distributions in the inner rail.

The first three axles of the locomotive exert a transverse force on the rail as the rail head shows compressive stresses on the inside (II/2) and tensile stresses on the outside (II/3). The transverse stress produced by the first axle is admittedly, small

by the stresses on the outside of the rail flange which are greater than under a predominantly vertical load. The stress distribution thus shows that the second and third axle of the locomotive foul the inner rail. The fourth axle exerts a force which is predominantly perpendicular to the rail tread, and its wheel flange does not foul the inner rail.

In conclusion, the stress distributions in the two rails of the curve show that the first axle of the locomotive fouls the outer rail, the second and third axles foul the inner rail, and the fourth axle rolls freely.

In examining the curves of figure 5, the reader may have noticed certain peculiarities in the stress variation. Among the most remarkable features are the very marked opposite peaks in the maximum compressive stresses of curve I/3. This phenomenon is also encountered in the following illustrations. It is characteristic for the stress fluctuation at certain well determined points, and therefore worthy of special attention. It has, in fact, also been noticed by other research workers, e.g. in the course of strain measurements on railway rails carried out by the Swiss engineer SCHLUMPF (3). But as the phenomenon has no immediate bearing on the appreciation of the running of vehicles through curves, its discussion may be deferred to section 6.

is concerned, of a modification carried out on the vehicle. By way of example, we may discuss the changes in the curve running qualities of a metre-gauge locomotive 1-E-1 consequent upon a modification of the truck axle drawback springs.

Figure 6 shows diagrammatically the axle distribution of this locomotive. The two truck axles are formed by Krauss-Helmholtz pony trucks (fig. 7) and have a transverse mobility of 150 mm on either side. The outermost coupled axles have a transverse play of 15 mm. The three other coupled axles are fixed, and the wheel flange of the central axle, which is the driving axle, has been dressed down by 13 mm.

The springing of the truck axles of the

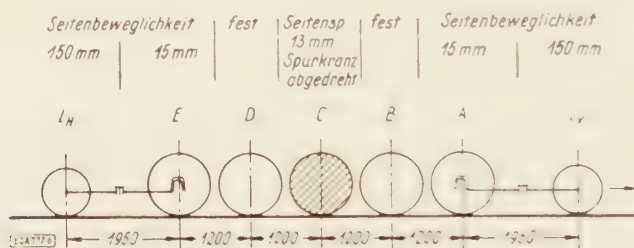


Fig. 6. — Axle arrangement of a narrow-gauge 1-E-1 metre gauge locomotive.

N. B. — Seitenbeweglichkeit — transverse mobility. — Fest = fixed. — Seitenspiel... transverse play 13 millimetres, wheel flange dressed down.

4. Change in the stress distribution in the rail due to a modification of the behaviour of the vehicle in the curve.

The stress distribution in the cross-sections of the rail in a curve not only conveys a picture of the behaviour of a given vehicle in the curve; it is also very sensitive to any modification of the running qualities, caused e.g. by structural alterations to the vehicle. Inversely, the change in the stress distribution permits an immediate assessment of the effectiveness, as far as the running through curves

Krauss-Helmholtz pony truck consists of leaf springs at the pivot and coil springs at the truck axles. The first running tests with the springing in its original condition showed that the original restoring springs were too stiff to allow the running through curves with a radius of 60 m or less. In curves with small radius, the rails were pushed apart so that the locomotive was derailed. Particularly stiff were the restoring springs at the pivot. It was necessary to improve the running through curves by reducing the force of these springs.

The effectiveness of this measure was checked by means of strain measurements in a curve of 57 m radius for the following three cases :

1. Pony truck with original truck springing;

2. Truck springs completely removed; this was done in order to ascertain the general influence of the spring on the behaviour in curves;

3. Truck axle springing in its original condition; pivot springing weakened.

the tensile stresses occurring on the outside of the outer rail (I/1, I/3, and II/3), and from the compressive stresses in the flange on the inside of the outer rail (I/2), that the two truck axles (L_H and L_V) as well as the two outermost coupled axles (E and A) exert considerable transverse forces on the outer rail. In other words, their wheel flanges foul the outer rail violently. The direction of the loads of the other axles is predominantly perpendicular to the rail tread. The con-

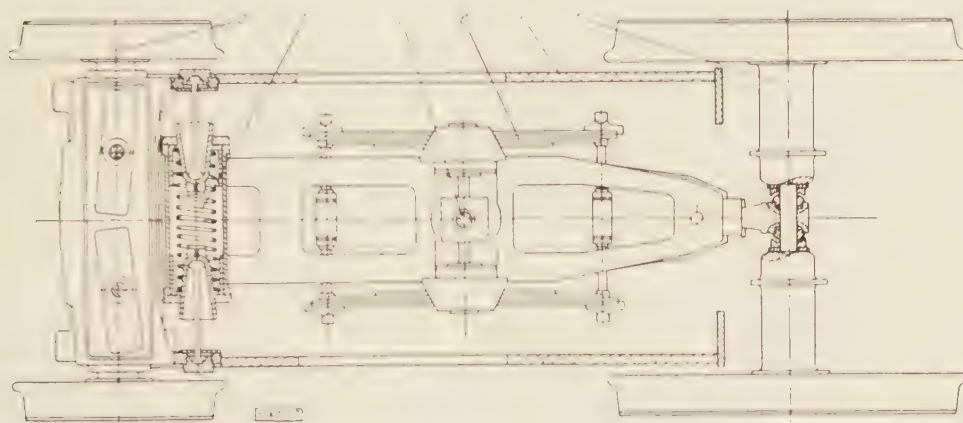


Fig. 7. — Krauss-Helmholtz pony truck.

- a) truck axle;
- b) truck axle coil springs;
- c) truck pivot (fixed to the frame);

- d) pivot leaf springs;
- e) frame;
- f) coupled axle.

These three cases were tested in three series of successive trials with the aid of strain measurements. The results of these measurements are shown in figures 8, 9 and 10.

In the course of these trials, the locomotive passed over the trial section at speeds ranging from 20 to 30 km/h. The influence of the speed has not been the subject of particular studies.

Figure 8 shows three charts with the stress distributions measured in the rail cross-sections under the influence of the locomotive with the original, very hard restoring springs. It will be seen from

siderable transverse force encountered under the truck axles and the outermost coupled axles is also apparent from the strong tensile stresses encountered on the outside of the rail flange (II/1). The flattened peaks are probably due to the measurement technique adopted. The strain gauge has here reached the limit of its range where the indication is no longer in linear proportion to the elongation.

The stress distribution in the inner rail shows that this rail is fouled by the wheel flanges of the three coupled axles in the centre. Under these axles, the rail head is subjected to tensile stresses on the out-

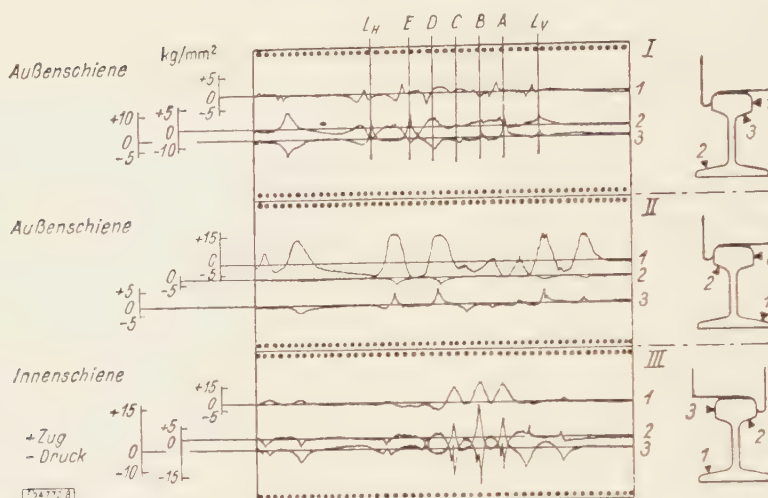


Fig. 8. — Stress distribution under a narrow-gauge locomotive 1-E-1 in a curve of 57 metres radius, with powerful return springs on the Krauss-Helmholtz pony truck.

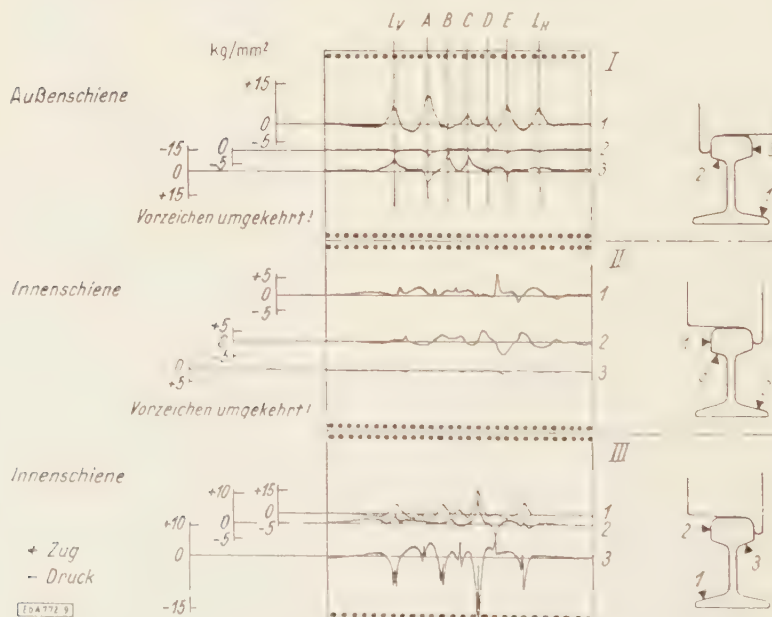


Fig. 9. — Stress distribution under a narrow-gauge locomotive 1-E-1 in a curve of 57 metres radius, without drawback springing on the Krauss-Helmholtz pony truck.

N. B. — Vorzeichen umgekehrt = sign inverted.

side (III 3) and to compressive stresses on the inside (III 2). The other axles exert a practically perpendicular force on the inner rail.

Entirely different are the stress distributions in the cross-sections of the rails when the restoring springs are entirely removed from the truck axles (fig. 9).

The first fact to be noticed is that the two truck axles now hardly exert any transverse forces on the outer rail. The rail head shows compressive stresses on

flange shows strong tensile stresses on the outside (III 1). The other axles exert a force practically perpendicular to the rail tread. The inner rail is apparently slightly fouled by the wheel flanges of the truck axles.

Figure 10 shows the stress distributions encountered in the third case where the return springs were restored to the pony truck but are weaker than they were before. The presence of the springs is immediately apparent from the transverse

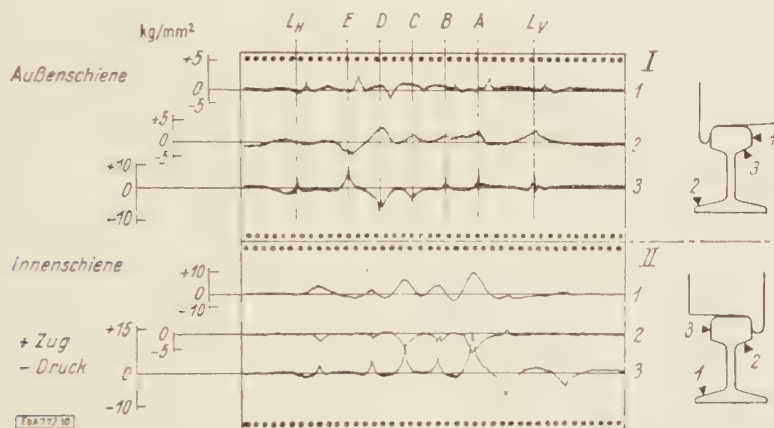


Fig. 10. — Stress distribution under a narrow-gauge locomotive 1-E-1 in a curve of 57 metres radius, with weakened drawback spring.

the outside (I.3) as well as on the inside (I.2), the flange shows a moderate tensile stress on the outside. Transverse forces are still encountered at the two outermost coupled axles. This result was to be expected as the side springs had in any case only been acting on the truck axles. The other axles behave as before.

There is also a change in the stress distribution at the inner rail. The transverse force is now concentrated at the fourth coupled axle, counted in the direction of running. The rail head is subjected to tensile stresses on the outside (III/2), and to strong compressive stresses on the inside (III/3), whilst the rail

forces which are now again exerted also by the truck axles, though to a lesser degree than with the original arrangement. The effect of the other axles remains the same. The outermost coupled axles again foul the outer rail. The load on the inner rail is about the same as in the case where the return springs were absent. The greatest transverse force is exerted by the fourth coupled axle.

To sum up, the stress distributions and their variations permit the following conclusions :

1) With the strong drawback springs of the original arrangement, the locomotive takes up a position symmetric to the

centre line of the track and exerts, with its truck axles and its two outermost coupled axles, a considerable transverse force on the outer rail, and with its three coupled axles in the centre a considerable transverse force against the inner rail;

2) When the return springing is removed, the two outermost coupled axles assume a guiding function and press against the outer rail. The inner rail receives pressure from the fourth coupled axle only. The position of the locomotive is no longer symmetric in relation to the centre line of the track;

3) With weakened springing, the truck axles press again against the outer rail, though with a weaker transverse force than in the first case. The position of the other axles is the same as in the case where the springs are removed.

The position of the locomotive on the track is therefore governed not only by geometrical data such as wheel base, transverse mobility and transverse play of the axles but also by mechanical data such as, in the present case, the force and distribution of the drawback springs of the pony trucks.

5. Future perspectives.

It is logical to attempt the determination, not only of the stress distribution in the rail, but also of the forces exerted by the vehicle on the rail which have caused this stress distribution. The stress distribution alone can, at best, only give an idea of the value of the bending moment in the rail cross-section concerned unless, under the impact of transverse forces, the rail is also subjected to torsional stresses. In this case, the measured normal stresses also contain portions due to torsion. In order to determine the forces acting on the rail, it is also necessary to possess information about the rail fastening conditions, about the length of the rail section deformed by an axle or several adjacent axles, about the ballast coefficient, about the friction between wheel and rail, etc.

This cannot be done without adopting certain simplifying assumptions. Mainly, however, it is necessary to obtain measurements of the deformations at a sufficiently large number of points of the rail section concerned so that adequate data for the determination of the forces can be procured.

It is intended to develop the method with these considerations in mind so that, in a comprehensive study of the behaviour of a railway vehicle in a curve, it should also be possible to determine the magnitude of the forces concerned as a function of the running speed.

6. Some peculiarities of stresses in rails.

In curve I 3 of figure 5, which shows the fluctuation of the stresses on the inside of the head of the outer rail, the stress maxima caused by the locomotive axles B, A, B are interrupted by very marked peaks in the opposite direction. For a brief moment, the predominant compressive stresses are overlaid by very considerable tensile stresses. The intensity of this superposition varies greatly. Under the front axle (B) of the locomotive, the initial compressive stress is reduced by more than half. Under the last axle, the rail head, otherwise subjected to compressive stresses, is for a very brief moment subjected to tensile stresses. This superposition is also very marked, under all the axles of the 1-E-1 locomotive, in the lower curve (III 3) of figure 8 which relates to the inner rail.

The same phenomena have been observed by the Swiss research worker SCHLUMPF (3) in the course of stress measurements on railway rails, carried out by means of strain gauges. He distinguishes between what he calls a primary stress which is due to the deflection of the rail as a whole under the influence of the wheel pressure and which we have called, earlier on, the predominant or initial stress, and a secondary stress which results from a

local deformation of the rail head. His interpretation of the phenomena, generally speaking, also explains certain peculiarities of the secondary stresses found by us. These peculiarities can be summed up as follows :

1. The secondary stress is always produced in the form of a tensile stress. It is therefore particularly obvious when it is superimposed on compressive stresses. If it is superimposed on tensile stresses, the relatively flat maximum of the primary stress is surmounted by a sharp peak. A superposition of this kind can be clearly seen in curve I/3 of figure 9;

2. This superposition is only observed with stresses on the underside of the rail head, but it is encountered without exception in all the curves relating to the underside of the railhead at the inner and outer rails, on the inside as well as outside;

3. As far as can be judged from the results of measurements obtained so far, the secondary stresses are not influenced by transverse forces. They arise irrespective of the fact whether the load is predominantly vertical or whether it contains transverse forces, although their intensity may vary slightly;

4. A more detailed examination also shows that the peaks of the secondary stresses are not symmetric in relation to the maxima of the primary stresses. In the inner rail, the peaks of the secondary stress generally precede the maxima of the primary stress. In the outer rail, it is the outer way round.

The peculiarities described under points 3 and 4 above are not yet proved with absolute certainty by the results of the measurements. Owing to the slow paper feed of the oscillograph, the maxima even of the primary stresses alone are relatively steep so that it is not possible to observe the shape and position of the secondary stresses with sufficient clarity. In future stress measurements at the rail, special attention will be paid to the secondary stresses.

However, the following interpretation of the origin of the secondary stresses appears plausible inasmuch as it accounts for the peculiarities of this superposition and is also compatible with the explanation given by SCHLUMPF.

The secondary stresses arise from a local deformation of the rail head immediately below the wheel in a zone of about 5 to 10 cm length. The extent of this zone can be estimated from the known width of the zone in which the primary stress is encountered, and from the width of the peak of the secondary stress. The rail head acts in this case as a self-contained beam, dissociated from the rail web and elastically supported by the latter. We are virtually confronted with two different elastic systems, simultaneously subjected to the same load.

One of these systems is the rail as a whole which is elastically supported by the sleepers and which, according to ZIMMERMANN (4), can be regarded with satisfactory approximation as a beam resting on a continuous elastic foundation. The application of the wheel load to this system gives rise to the primary stress distribution in the rail. The magnitude and distribution of this stress depends not only on the wheel load but also on the factor

$$L = \sqrt[4]{\frac{EJ}{bC}} \text{ or,}$$

in other words, mainly on the ratio of the bending resistance EJ of the entire rail and the modulus of elasticity bC of the support (where C is the ballast coefficient in kg/cm^3 , and b the effective width of the ballast in centimetres). For the rail as a whole, L is approx. 100 ± 10 cm. The stress produced by a concentrated load without adjacent loads is distributed symmetrically in respect of the load application point over a rail section of the length

$$l = L\pi,$$

which is thus proportional to the factor L .

In the case of the other system, that of the rail head on elastic supports, it must first of all be noted that the bending resistance EJ of the rail head is markedly lower than that of the rail as a whole. With the S 49 rail, the bending resistance of the rail head is only one-fiftieth of the rail as a whole. On the other hand, the modulus of elasticity of the rail web is greater than that of ballast and sleepers so that the factor L for the support of the rail head is considerably smaller than the factor L for the support of the rail as a whole. As has been demonstrated earlier on, the decrease of this factor is accompanied by a decrease in the extent of the zone over which the stress is distributed in the rail head. From the width of the stresses measured, the factor L for the rail head is estimated to amount to between 1/20th and 1/30th of that of the rail as a whole. The stress maximum is also directly proportional to this factor so that a smaller factor L is also matched by a smaller stress.



Fig. 11. — Bulging of the rail head ahead of the moving wheel.

But the measurements show that the peaks of the secondary stresses are far from being as much smaller as would correspond to the ratio of the widths of the two stress maxima so that it must be assumed that the secondary stresses are also due to other causes.

It is conceivable that there is also a local deformation of the rail head due to the fact that the rail bulges slightly immediately ahead of the wheel, rather like the dough under a rolling pin. In figure 11, this phenomenon is illustrated

in a slightly exaggerated form. The deformation of the rail head is of course also imparted to the rail web. But due to the elasticity of the latter, the deformation is spread sideways and decreases with the distance from the rail head so that, at the rail flange, the local deformation almost ceases to be noticeable.

This spreading of a deformation between rail head and rail flange is also apparent from all the curves of the stress measurements. The peaks of the primary stresses in the rail head are always more or less sharp whilst those in the rail flange are rounded.

The non-symmetrical position of the secondary peak in relation to the primary peak, mentioned in Section 4 above, is probably due to the difference in the direction of the longitudinal friction between wheel and rail as the vehicle negotiates the curve. If the wheels are wedged on the axles so that they are forced to roll, the wheel rolling on the outer rail must cover a greater distance and that rolling on the inner rail a shorter distance than the distance which corresponds to the rolling of the axle. In consequence, the wheel slides forward on the outer rail and exerts on the rail a force in the forward direction. On the long rail, the sliding friction gives rise to a force directed towards the rear.

If this explanation is correct, the non-symmetry should no longer be observed, on straight or curved track, with wheels loosely fitted to the axles. As already mentioned it is now necessary to examine these phenomena more closely by means of appropriate measurements, as we intend to do.

Summary.

The present article deals with a method of investigating the running of railway vehicles in curves by means of strain measurements on the rail. The measurements are based on the distribution of the normal stress in one cross-section of the outer rail and inner rail and, where

necessary, on that in several cross-sections. From the distribution of the stress in magnitude and direction, it is possible to conclude whether, as the different axles of the vehicle pass the measuring point, the rail is predominantly subjected to vertical stresses or whether there is, in addition to the axle load, a transverse force due to the interaction between wheel flange and rail. The stresses are determined by measuring the strains on the surface of the rail, viz. on the side and underside of the rail head, as well as on the rail flange on either side of the web. The measurements are obtained electrically by means of inductance detectors. The indication is made visible and recorded by means of a loop oscillograph. This recording, which is almost without inertia, also permits the measurement of rapidly fluctuating stresses so that it is also possible to ascertain the behaviour of vehicles negotiating a curve at high speed. An example shows how it is possible to use this method also for the purpose of ascertaining the effect of structural alterations to the vehicle. It is thus possible to verify the effectiveness of structural measures taken to improve the running of railway vehicles through curves.

In the present article, the running of vehicles through curves is examined on

the basis of a purely qualitative discussion of the stress distribution. But the final objective is to make it possible, with the aid of such stress measurements in conjunction with deformation measurements, also to determine the magnitude and direction of the forces exerted by the vehicle on the rail. In conclusion, the author discusses certain peculiarities of the stress distribution and stress fluctuation, which are not immediately characteristic for the behaviour of the vehicle in curves, but which are very remarkable.

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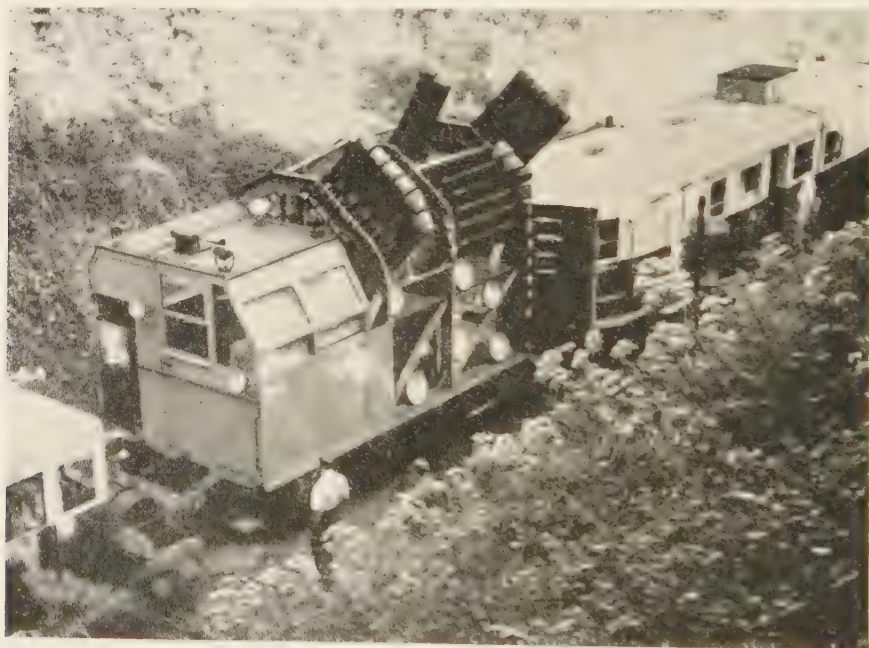
French Railways tunnel-gauging apparatus.

Continuous graph of tunnel profiles produced by pens worked automatically by roller-fitted feeler-arms.

(The Railway Gazette, March 1, 1957.)

The French National Railways, early last year, placed in service a second apparatus for measuring tunnel contours. This is known as a « Palpeur-mesureur Castan » or Castan « feeler-measurer »

a number of rollers carried at the ends of a series of arms. These roll along the tunnel walls as the apparatus moves along the track and their undulations are used to influence pens which register the varia-



Castan apparatus showing feeler-arms extended and arrangement of floodlights.

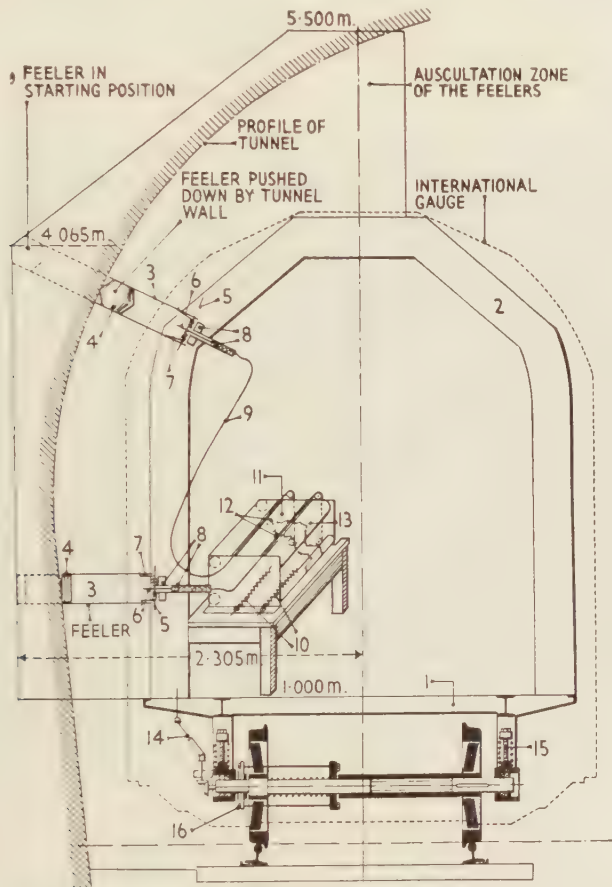
apparatus. The general principle of the new apparatus remains the same as that used in the original model, which has been in use since 1940, but there have been some improvement in details and the dimensions are larger. The apparatus has

tions on a continuous moving paper band. This is a principle not previously used for this work.

In our issue of April 2, 1954, a description was given of a « clearance car » used by the Pennsylvania Railroad for

measuring tunnel profiles and the clearance of lineside structures. This car also uses feelers, or antennae, but they are used with the train standing still, and it

a system of sliding rules is used which enables isolated profiles to be read rapidly, and the Eastern Region of British Railways has developed an apparatus using a



Sectional diagram of car showing connections from arms to recording apparatus.

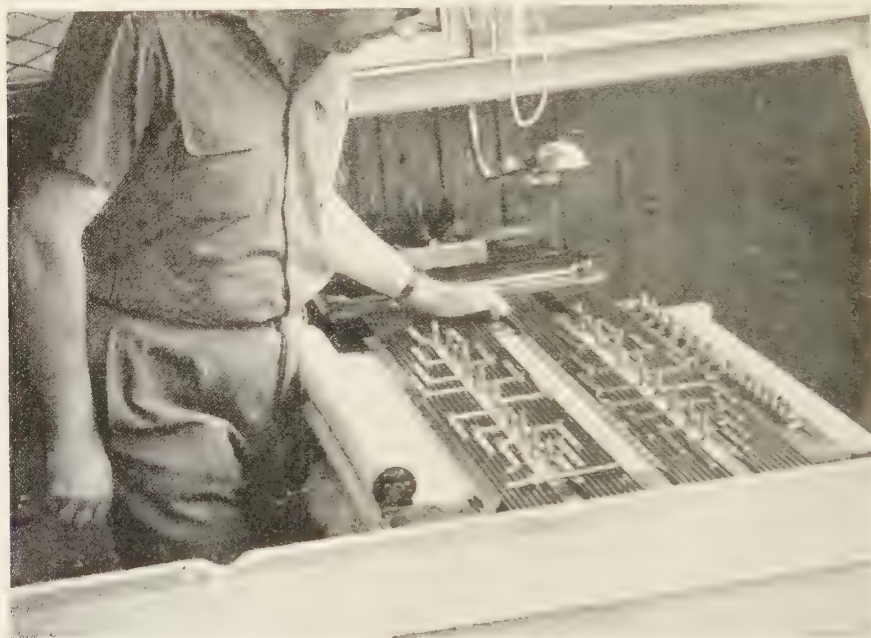
- | | |
|------------------------------|--------------------------------------|
| 1. Platform. | 10. Cable-end springs. |
| 2. Frame. | 11. Recording apparatus. |
| 3. Feeler-arms. | 12. Guides and pens. |
| 4. Roller. | 13. Paper-feed cylinder. |
| 5. Axis of rotation. | 14. Drive to recording apparatus. |
| 6. Cam. | 15. Platform-carrying apparatus. |
| 7. Feeler-arm return spring. | 16. Distancing apparatus for wheels. |
| 8. Pad and slide. | |
| 9. Bowden cable. | |

is necessary to read the position of each feeler on a scale inside the car when each profile is taken. The result is a series of graphs of isolated profiles. In Germany,

single telescopic arm which allows clearances to be established with some speed. The British Railways apparatus was described in our issue of October 22, 1954.

The Castan apparatus does more than establish isolated profiles. It registers the profile for the entire length of a tunnel, bridge, or other obstacle, and by inspection of the graphs the narrow sections of tunnels, low or narrow portions of arches, projecting points of rock, or structures too near the line can be recognis-

are capable of being extended during operations to ensure that the wheel flanges are pressed firmly against the rails to avoid any side movements of the wagon on the track. The sliding arrangement is applied to the rail on the left, looking in the direction of forward movement of the vehicle. When operations have been com-



Recording apparatus showing pens, guides, and paper cylinders.

ed at once. The profiles concerned are produced on the graph on a scale of 1:10 in relation to the movement of the rollers on the walls of tunnels or over the surface of other structures.

The basis of the apparatus is a two-axle wagon on which is mounted an arrangement of three frames carrying the feeler-arms. It also mounts a cabin which houses the registering apparatus. The wagon measures 24 ft. 7 in. over buffers and is 9 ft. 4 in. in width. It has a strongly-braced rigid frame. The axles

pleted, a keying system enables the wheels to be locked at the normal gauge for travelling purposes.

The floor of the wagon is carried by the axles through spiral springs fitted between guards. The springing can be completely suppressed by a screw apparatus during operations so that the platform is carried at a fixed distance of one metre above the plane of the track. The springing is restored when work is completed and the wagon can then be allowed to travel at an average speed of 25 m.p.h.

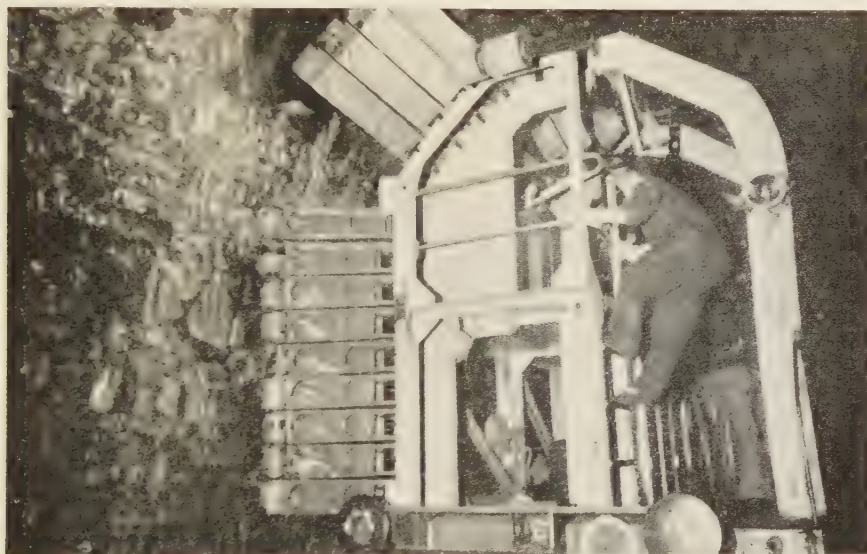
The positions adopted for the spindles carrying the obstacle-measuring apparatus are well clear of the track.

Feeler-arms.

The three frames, on which are hinged 31 feeler-arms, constitute a rigid metal structure which represents the gauge limit of the apparatus itself. This contour is

studied for electrification and have a margin of 5 cm laterally beyond that required for metal bridges. The six other arms in the working zone are intended to supplement the number of contact points where serrated arches are encountered.

Each feeler-arm consists in essence of a hollow metal box, a roller and a cam mounted on a crossbar. These three



Apparatus at work in a tunnel with feeler-arms pressed down by the walls.

inscribed within the minimum gauge used by the S. N. C. F. with a horizontal margin of 5 cm and a vertical margin of 3 cm. As this minimum gauge is smaller than that for « transit vehicles » — the international gauge — the apparatus can circulate anywhere on lines open for vehicles used for cross-frontier trains. The 31 feeler-arms are divided between the three sides of the frame in a manner which assures that they will function whatever the degree to which they are pressed down by the obstacles encountered. The first 25 arms cover a zone which suffices for the largest gauge used in

elements are firmly connected. The whole box, which is in parallelepiped form, is reinforced by cover plates and the corners are also reinforced to keep them in shape. At one end is a shock-resisting roller in light alloy with a hard steel rim. The roller is mounted on a steel axle and is assembled so that it is held by the end of the box between two shallow bowls containing ball races.

The other end of the box, which has a light-alloy crossbar, carries the cam and is attached to the frame by a ball-bearing mounting. The cam is a most important piece of apparatus as it transmits the

measurements taken of the obstacles. Its movement is a function of the length of the arms. It is made of hard steel with a cemented edge. When an arm encounters an obstacle it rotates about its pivot as it is pressed down, being held in contact with the surface of the obstacle by springs. As it moves, the cam moves with it.

The cam in turn transfers the movement to the recording pen on a scale of 1:10. This process gives the distance of the obstacle in relation to the fixed parts of the apparatus.

The remainder of the apparatus transmits the measurements taken by the cams to the recording apparatus. For that purpose, the cam carries a steel roller mounted in ball-bearings on a slide lying in a grooved pad. This is maintained in constant contact with the cam by a coil spring. To the roller is attached a Bowden cable. The cable is guided by pulleys and leads to the recording apparatus. For movement of the wagon when not working, the arms are folded inside the frames and held in place by locking points provided on the cams.

Recording machinery.

The recording apparatus has 47 pens, including 16 which are fixed and which

trace the parallel reference lines. The remaining 31 pens are connected through a pen-carrying apparatus to the 31 cables, one to each arm. The pens mark on a metal table measuring 2 ft. 6 in. and 1 ft. 8 1/2 in. They are guided by rules and leave a track in the direction of movement of the paper over the table. The paper, 2 ft. wide, is moved by mechanism worked from the front axle. It moves at either 2 mm or 5 mm per metre of track covered by the vehicle.

The recording apparatus is enclosed in a glass and metal cabin measuring 9 ft. 4 in. x 6 ft. 6 in. This has wood cladding and is insulated by glass wool to give protection to the operators from bad weather and to prevent the ink from freezing. As the apparatus is used principally in the mountains this precaution is essential.

This second Castan apparatus has been in service less than a year but has already dealt with 80 tunnels and covered some 5 000 miles of track. It has also measured some 300 bridges. The original Castan apparatus has run some 50 000 miles in France and in Luxembourg, Germany, and Austria, and has measured more than 2 000 tunnels. After some reconstruction, particularly of the feeler-arms, it will be ready for further service.

The case for the shorter freight train,

by E. HARRISON RIGG, (*)

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—

Maybe it's false economy to run freight trains too long. A railroad analyst comes up with a new approach to this question, using a factor that takes into account value of freight carried and time it takes to move to destination.

Railroads are paid for moving *cars* — not trains.

Whether a shipper uses the railroad or turns to a competitor depends, in most cases, on the performance of his cars from loading point to consignee — NOT on the railroad's performance figures for trains.

Thus, car performance and the cost of moving cars should be prime concerns of railroad management. (By car performance, we simply mean scheduled time from shipper to consignee — or origin to destination for empty cars.) Yet practically all railroad control statistics — gross ton miles per train hour and the like — are based on the performance of *trains*. Years ago, under monopoly conditions, such statistics aided the engineering progress of the railroads. Today, however, the railroads have strong competition. Conditions have changed, but the statistics have not.

There is no way to find the performance figures for *cars* from the performance figures for *trains*. Average train speed could be increasing, but at the same time car speed could be decreasing. Train schedule performance could be improving while car schedules deteriorate. It is possible to improve train performance statistics while ruining car performance with undue « waiting for tonnage », decreasing frequency of local service, and similar practices.

There is a definite relationship between the number of freight cars in a train and the costs of moving freight. From such a relationship, it is easy to calculate what length of train will move freight for the least costs. In doing that, however, we will consider the costs of *moving freight* in addition to trains.

What are the costs of moving freight? The obvious ones are well recorded — engines, car equipment, yards, tracks, train and engine crews, clerks, maintenance employees and supervision. These are included under operating expenses.

Are there other costs? Yes, there is the « per diem » cost of the empty car, at \$2.40 a day. We can reliably figure 10 cents per hour for the shell of a car — loaded or empty.

But there is still another, and very important, cost factor. Suppose a car is loaded and a \$500 freight bill paid to the railroad. If this car is moved to destination, the \$500 is earned. If the railroad fails to move the car at all, then it must return the \$500 to the shipper — and the cost of *not having moved the car* is \$500 (minus out-of-pocket savings).

If the railroad moves one of these cars daily, it receives \$3500 per week, against which are charged out-of-pocket costs, leaving a net gain in revenue. However, if performance isn't satisfactory, there will

(*) « With the modern trend of thinking, maybe the time is ripe to examine the subject of the length of freight trains generally », says the author of this article. Mr. Rigg has had 20 years of operating experience with the Long Island and Pennsylvania Railroads. Prior to his present position he has served as Transportation Apprentice, Assistant Yardmaster, Trainmaster and Supervisor of Safety.

be a decrease in the number of cars moved (some will be lost to competition). For each car lost there is a decrease in the net gain in revenue, or a *cost chargeable to the poor performance*.

If these cars require five days from shipper to consignee, it is simple arithmetic to charge so much per hour « cost » for not moving a car at all, for the hours the car is idle resulting in bad performance, or for each hour the car is moving. When moving, there is a value per hour being *earned* by the car; if not moving, so much cost per hour is charged against the car. (Moving or not, as long as it gets to destination the car earns a certain value per hour, and also consumes a certain cost per hour, if earnings and costs are averaged over the total time consumed in earning the amount of the freight bill.)

We know that if the cars do not move there is the greatest cost offsetting the net gain in revenue; if they move with good performance there is the least cost to be subtracted from the net revenue gain; and if there are bad performances the effect on net revenue gain varies between these extremes. The logic is exactly similar to the 10 cents per hour charged to the shell of the car, but the value per hour charged to the load is in addition to the 10 cents and will generally be much more.

Thus, there are three general costs for moving freight: (1) the per diem cost for the shell of the car; (2) the « per diem cost » for the loaded freight; (3) the operating costs to move the car physically.

If you doubt that these three cost items are present in moving freight, consider the over-all « per diem values » of empty cars, of carloads of cinders, of carloads of fresh meats. Under competitive conditions, there is a big difference in value per hour that such cars cost if they do not move (under monopoly conditions the values, roughly speaking, would approach equality).

Just what is the relationship between the number of freight cars in a train and the costs of moving freight?

To simplify the formulas that follow, we

will combine the first two above costs, calling it the « cost per hour per car and load », and we will assume that the third cost varies with the number of train and engine crews used per car. This assumption is not far from the actual facts. For example, many trains with two or more units of power can be split in two, and the only material increased costs would be crews. (Also, with two trains instead of one, there is one-half the probability that a defective car will stop the movement of the cars, and so on.)

As the number of cars not moving increases, « car and load costs » increase, and as more trains are run, « train and engine crew costs » increase and vice versa. Also, if we wait to accumulate more cars per train, then « car and load costs » increase, but « train and engine crew costs » decrease.

Since these cost values vary regularly, there is an « in-between » value that will produce the *least over-all costs*.

Consider any terminal originating trains it:

A = cents per hour costs per « car and load »;

N = number of cars in trains;

R = hourly rate of cars accumulating to move in trains of all classifications;

K = number of different track classifications taken by trains from the yard;

C = cost per train and engine crew used (cents);

T = total costs of « car and load » and « crews » costs per day.

Then:

$$T = 12 \times A \times K \times N + \frac{24 \times R \times C}{N}$$

The first term on the right, $12 \times A \times K \times N$, equals car and load costs per day (irrespective of the number of cars handled). The second term on the right, $\frac{24 \times R \times C}{N}$,

equals train and engine crew costs per day.

We can see from the formula above that as N increases, « car and load costs » increase, but « train and engine crew costs » decrease. As N decreases, « car and load costs » decrease, but « train and engine crew costs » increase. There is a value of N which will produce the least total costs. That value is:

$$N = \frac{\sqrt{2 \times R \times C}}{A \times K}$$

Let's take a couple of examples of this relationship. Suppose $R = 100$ cars per hour; $K = 15$ and $C = \$100$ or 10 000 cents. If $A = 53$ cents (representing a high-value commodity), then N — the length of train to produce the least total costs — will be 50 cars.

However, if A is only 13 cents (representing a low-value commodity) then the ideal length of train, N , would become 100 cars.

In the second formula, the R , C and K are easily determined. The A is more difficult. Traffic Departments can place a value per hour on each general commodity, to figure the allowable costs that can be incurred to move that commodity (see explanation at the end of this article).

Basically, the value of N in the second formula is the defense for short, « hot-shot » trains, because the value of the freight makes A a sizable amount.

There would seem to be many more possibilities for shorter freight trains. For instance, if the value lost by not moving a certain commodity is \$1 per hour per car and 50 such cars have been waiting eight hours to move, a cost item of \$400 is involved. The physical costs of moving the cars, assuming multiple power units, are practically the same in a 150-car train as in a 75-car train — or a 50-car train — except for engine and train crews. And so crew costs need only be matched with the car value costs.

Much of our present thinking in yard and terminal operation takes cognizance only of a value of 10 cents per hour per car — whereas actually this value should be increased many times when dealing with

loaded cars. The per diem cost of cars accumulating in yards is amazing. In many cases, *even at 10 cents per hour per car*, the total becomes larger than all the wages paid to operate the yard!

Are we emphasizing « gross tons per train » too much? Or should we start paying more attention to *carload* costs?

How to determine « K » and « A ».

« K » is the number of different track classification combinations taken by trains from a yard as a general procedure. Where two combinations use a common track, it is that arrangement of combinations which is the least number for which all tracks will be included.

If number of track classifications is.	Names of tracks.	Combinations taken by trains.	Then, K equals.
1	A	A	1
2	A, B	A, B, AB	2 1
3	A, B, C	ABC AB, and C A, B, C AB, BC and A, B, C	1 2 3 2

Etc.

« A » can be determined in several different ways, depending on the specific situation being studied. To take just one example:

Consider carloads of meat, Chicago to New York. If a schedule of longer than 72 hours would lose the business to competitors, then in 72 hours the freight revenue must be earned; i.e., the transportation produced. If the freight is \$360 per car, then each car is worth \$5 per hour if it does not move, minus the out-of-pocket costs of moving the car which are saved.

Operating ratio is a general gage of these out-of-pocket costs. So we could say, if our operating ratio is 80, that the net loss is 20 % — or \$1 per hour — by not moving the car. Thus, in this case « A » would be \$1.10 — \$1 for the load of meat plus 10 cents for the « shell » of the car.

Let's look at these four-wheelers.

(*Railway Locomotives and Cars*, February 1957.)

Rock Island is now operating nearly 50 « Convert-A-Frate » cars arranged for handling demountable cargo units; others of these ACF Adaptos are under test in « Truc-Train » service on Pennsy.

ACF Adapto is on its way on the Rock Island. By the end of the year, the road was operating 43 of these cars in revenue service. These RI cars are arranged to handle the removable gondola and box

service on numerous American railroads. There seems to be little doubt that this newest fleet is the largest group of four-wheelers built and successfully operated in revenue service here in many years.



Rock Island ordered fifty Adapto flat cars. Containers delivered included sixty 17-ft boxes, twenty 17-ft gondolas, and ten 35-ft boxes.

units which were also supplied by ACF. After delivery, the Rock Island built automobile carriers for the cars, and encouraged the construction of a refrigerated box which is now in use. These Adapto cars are also being tested in piggy-back trains on the Pennsylvania.

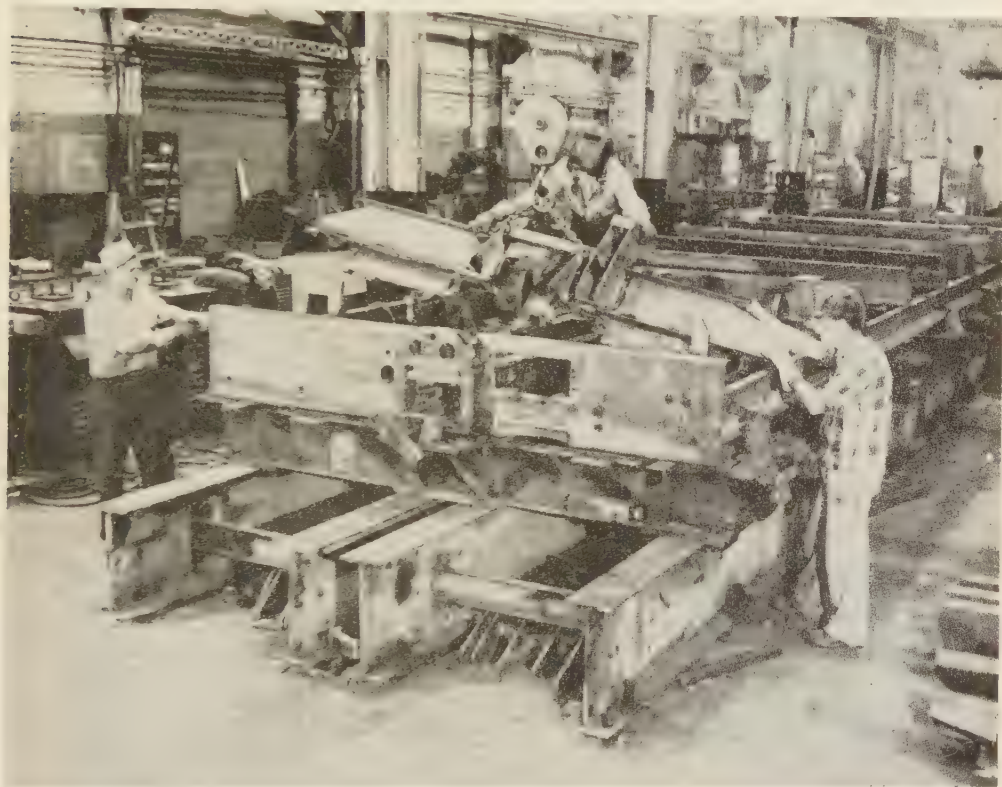
The car itself is a radical departure. Until nearly the turn of this century, there were many four-wheel coal cars in

The center sill for this car is composed of two 41.2-lb AAR Z-26 sections extending the full length of the car. This sill is reinforced along its center 12 ft. with a $\frac{3}{8} \times 5$ -in. bar welded under the joint between the two Z sections. A $\frac{3}{8} \times 2\frac{1}{2}$ -in. bar is welded on the top surface of each of the bottom flanges of the center sill for the same 12-ft. length.

Side sills are 9-in., 13.4 lb LAHT chan-

nels which are connected to the center sill by eight 10-in., 15.3-lb channel cross bearers and by two $\frac{1}{4}$ -in. pressed end sills 19-in. deep. There are eight diagonal braces made of the same channels which are used for the side sills. These braces

of the car, but only out from each side of the center sill. The top of the decking is flush with the top of the center sill. It is attached to angles on each side of the center sill, to channel-type floor stringers, and to the tops of the side sill



Cross bearers are channels which run under the side and center sills.
Jig holds inverted underframe during assembly.

form an «X» in each of the floor panels between the cross bearers just inboard of the trucks. The entire underframe is a welded assembly.

Self-clearing floor.

The flooring is $2\frac{3}{4} \times 5\frac{1}{2}$ -in. treated decking spaced to be self clearing. This decking does not extend the full width

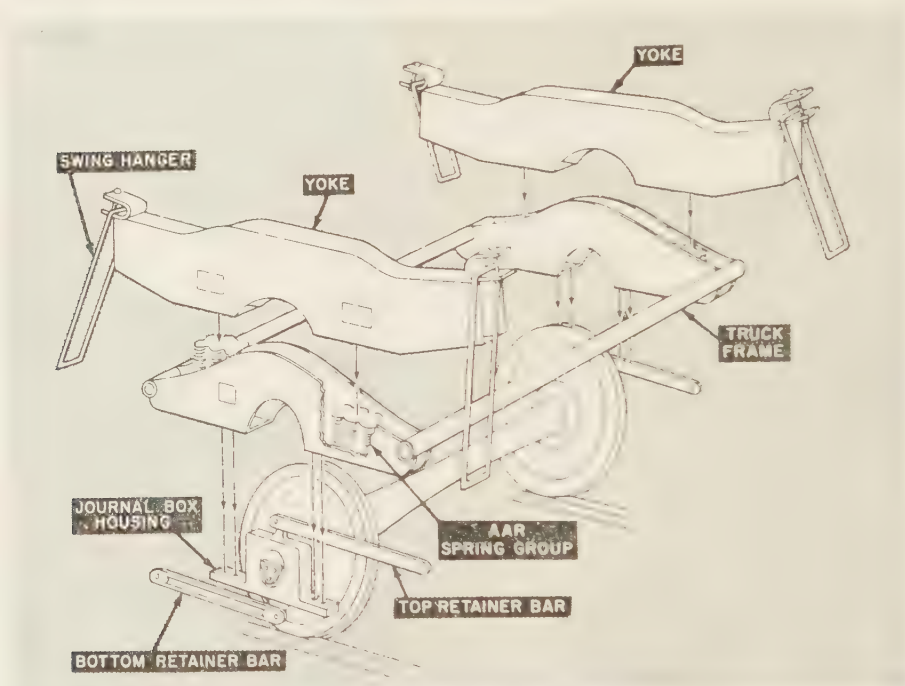
channels with $\frac{1}{2}$ -in. Nelson end-welded studs and watertight nuts. At the four points where the forks of the lift trucks contact this platform in unloading the 17-ft. containers, the deck is floored with pressed steel channels instead of the wood used over the remainder of the surface.

The Waugh twin-cushion draft gears are applied in the conventional manner

at the ends of the center sill. Forged rear draft lugs are riveted into the sill, while the combined striker and front draft lug assemblies are welded in at the ends. The cars have bottom-operated, Type F, interlocking couplers. The top of the car's deck is just above the top of the couplers, just $41\frac{7}{16}$ -in. over the rail.

make the Adapto truck self centering while permitting some lateral motion between it and the body.

The Rock Island cars have AAR standard 6×11 -in. Bethlehem axles and 33-in. Southern one-wear cast-steel wheels. Axles are equipped with Timken AP roller bearings. Each roller bearing



Single-axle truck supports the car body on four swing hangers as illustrated in this Railway Locomotives and Cars sketch.

Adapto truck.

The Adapto single-axle trucks applied to these cars are connected to the body through swing hangers supported at brackets on the cross bearers. This swing-hanger suspension is a departure from the arrangement previously used on American four-wheelers which have usually had their two single axles guided in pedestals fixed to the car body. The loop-type swing hangers are inclined to

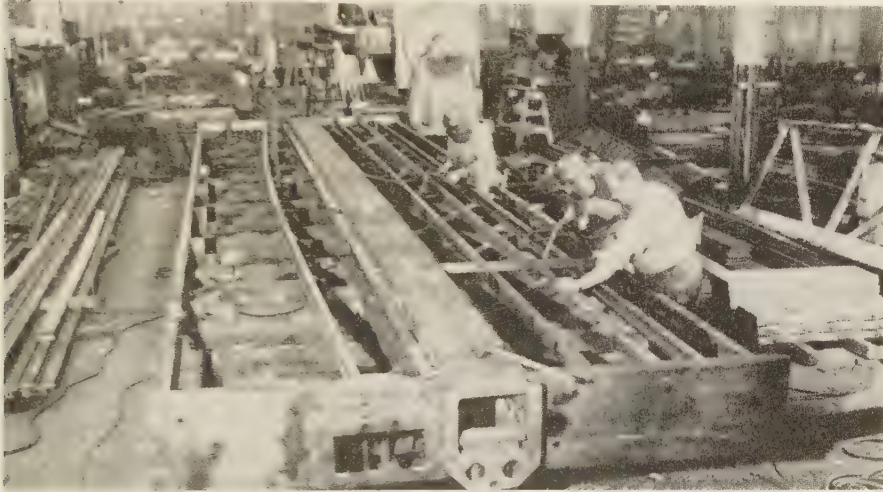
assembly is carried in a cast steel housing under a Fabreeca pad.

Each truck frame is composed of two side frames connected by two 4-in. steel pipes. The journal box housings are bolted into the side frames which are pressed from $\frac{5}{16}$ -in. plate and are shaped to provide space for two groups of standard AAR coil springs — one group on each side of the journal housing. This makes a single unit out of the wheel and

axle set, the journal bearings and housings, and the truck frame.

The coil spring suspension of each truck is four nests of five coils each. These

has the truck swing hanger brackets applied at each end. This yoke also has the bracket for the transverse yoke retaining bars.



Three floor (above) stringers support the wood decking between the center and side sill on each side of the car.



Another type Adapto container on flat car.

are the AAR 1947 alternate standard design springs with $3\frac{11}{16}$ -in. travel. Resting on the two spring groups on each side frame is a $\frac{1}{4}$ -in. steel pressed yoke which

The axles are kept at right angles to the car's longitudinal center line through the upper and lower retainer bars applied to the truck side frame. The lower bar

on each side is fastened to the end cross bearer of the body, and the upper bar is pinned to an intermediate body cross bearer. These yokes are kept in lateral alignment within the side frames through yoke retaining bars attached to the yoke and to the tubular side frame connecting members. All of the pins in these aligning mechanisms are carried in rubber-mounted bushings requiring no lubrication. Attached to the inboard side-frame connecting member of each truck are two Monroe, strut-type hydraulic shock absorbers which are then fastened to the adjacent body cross bearer.

Brake system.

The Rock Island cars have Westinghouse AC brake equipment actuating individual truck-mounted cylinders on each wheel. Each cylinder then operates a single Cobra brake shoe against the wheel through a lever pivoted at the side-frame connecting member. Despite the individual cylinders, the cars have Creco $\frac{1}{2}$ 18 brake beams, loop-type brake hangers, and the conventional wear plates and safety supports.

The light weight of the car is 27 000 lb. The length over end sills is 38-ft., the width over side sills is 9-ft. 4-in., and the wheel base is 27-ft. The capacity is 70 000 lb and the load limit is 77 900 lb.

In addition to its containers, ACF has

been developing high-speed anchoring mechanisms to attach these demountable bodies to the car. Also developed is a retractable trailer hitch for cars assigned to piggy-back service.

Rock Island experience.

When the Rock Island first put its « Convert-A-Frates » into service, they were hauled near the rear of trains. Today these four-wheelers are handled at any location throughout a train. The road has experienced no derailments, and imposes only a normal speed restriction when moving them through sharp turnouts. There have been no tests made with these cars in hump classification yards.

Rock Island test department studies have shown good vertical ride characteristics. The cars are handling a twice-weekly meat run from St. Joseph to Chicago. This meat is handled from the Swift & Co. plant in the plastic refrigerated unit equipped with meat hooks. There have been no cases when meat tore through the hooks — an indication of the riding qualities.

Rock Island inspections have shown the welded truck assembly has been satisfactory; a point of concern when the cars were first delivered. From a mechanical standpoint the entire truck was found to be in excellent condition.

Experiments with points and crossings.

German Federal Railway trials with partially- and fully-welded frogs.

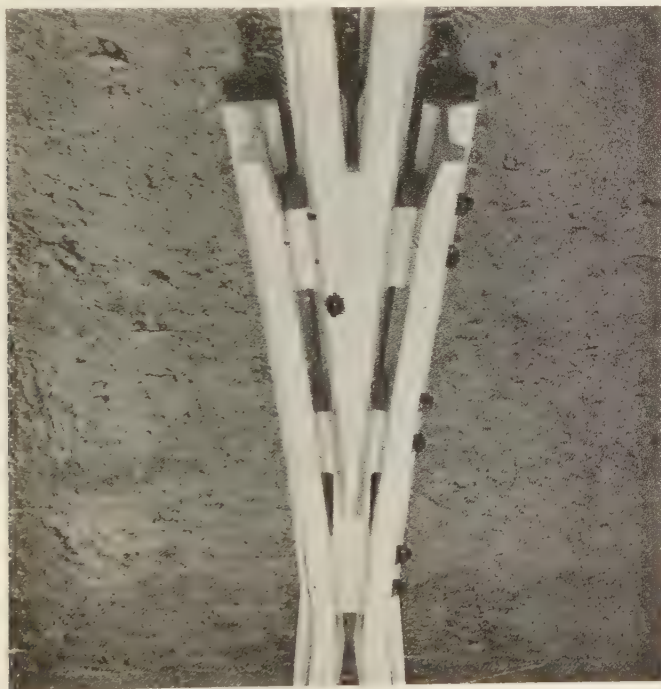
(*The Railway Gazette*, March, 29, 1957.)

A high traffic density, together with high speeds, causes considerable wear to points and crossings because of the successive hammer blows to which they are continually subjected. Consequently, it is necessary to carry out periodic inspection to points and

frogs, by Pintsch Bamag A.G., Butzbach, Hessen.

Design features.

The partially welded type of frog, in which the only loose parts are the through



Partially-welded crossing with bolted wing rails only.

crossings to ensure there are no loose parts such as distance pieces, bolts and nuts, and so on. Economy in permanent way maintenance is claimed by the developing and patenting of partially, and fully welded

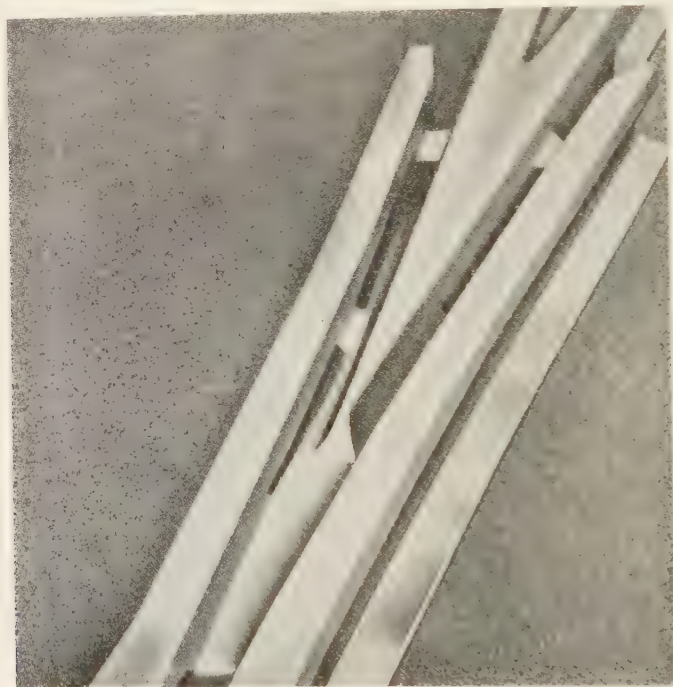
securing bolts and nuts has been installed on the German Federal Railway, with, it is said, considerable success. As much as 80 per cent saving in maintenance is claimed, compared with the ordinary build-

up type. Trials have also been conducted on the railway for some time with the fully welded frog, and from reports, are stated to be most satisfactory.

In both types of frogs the spliced point rails are welded together to form one piece, and all parts subjected to hammer blows have been specially hardened, care

which is welded to the point rails by flash welding. Special electrodes, as well as the treatment during and after welding, are necessary to ensure the bonding of the various components.

Care is taken during welding to avoid warping or cracking of the frog block, and final machining after welding ensures abso-



Fully-welded frog and wing rails after final treatment.

being taken to control the hardening process, so that it is gradually reduced to the normal hardness of the running rail; hardening is done by a special process which gives a maximum value of 120-130 kg/mm².

In both designs frogs are made up of normal rail sections. That part of the frog forming the distance piece and the nose at the point rail is made up from a forging

lute accuracy. Production costs are, it is stated, higher than those of the conventional frog, but it amounts to approximately half those for cast manganese steel frogs. It is claimed that the life of welded frogs is twice that of the normal type, but cost of maintenance is almost negligible, furthermore gauge widening is eliminated. Building up of the frog by welding, after wear gives good results.

Welded aluminium footbridge.

(*Engineering*, March 22, 1957.)

Shown in the accompanying illustration is an aluminium footbridge that has recently been completed for the Northern Aluminium Company, Limited, Banbury, over the River Ebbw, between the South Wales towns of Rogerstone and Bassaleg. The

The bridge is of through-type W-latticed girder construction, the top chord of each girder serving as a handrail. This top chord is curved to a radius of 226 ft., the depth of girder being 4 ft. 6 in. at the ends, and 6 ft. 3 in. at midspan. The 4 ft. 6 in.



This 60 ft. aluminium alloy footbridge is one of the first examples of all-welded construction for this type structure. The natural corrosion-resistance of the alloy enables it to be left unpainted.

bridge, which is of 60 ft. span, is one of the first examples of all-welded fabrication for a structure of this type.

The approaches carrying the pathway were limited to a gradient of 1 in 10, and the bridge was required to have a clearance, above the highest water mark, of 3 ft. These conditions meant that, if the approaches were not to be unduly long, the bottom chord of the bridge girders could not be lower than the points of support on each bank; this allowed little depth for bracing.

wide footway is similarly curved to give a 4 ft. 6 in. handrail height throughout. All the essential lateral rigidity is provided by transverse and diagonal bracing, in both horizontal and oblique planes, below the level of the flooring.

The bridge was designed for a live load of 100 lb. per sq. ft. (on a 15 lb. per sq. ft. dead load), with provision for a point load of 500 lb. Lateral design loadings were 50 lb. per ft. run on the handrail, and a wind load of 12 lb. per sq. ft. The bridge is in a reasonably sheltered position.

In order to use the structural material to best advantage, maximum use was made of bulbed and lipped extruded sections. The bottom chord of each bridge girder is a 4 in. by 3 in. by 1/4 in. extruded T, and the top chord and handrail a T with its flanges radiused and bulbed to provide a comfortable hand-grip. Lattice members are of 3 1/2 in. by 1 3/4 in. lipped channel section, and bracing members of 1 3/4 in. equal bulbed angle. The alloy chosen for all structural parts was Noral B51S (B.S. 1476 : HE30), fully heat-treated. In this condition, this alloy has a guaranteed proof stress of 16 tons per sq. in.; it has a high resistance to atmospheric attack, and has come to be recognised as the most suitable aluminium alloy for normal structural applications.

In welding a heat-treated aluminium alloy such as HE30, there is an inevitable local reduction of properties close to the weld though, after welding, the quenching effect of the adjacent metal usually brings about a partial recovery of strength. By avoiding transverse welds, the local loss of strength can be minimised; in this foot-bridge the usual practice of making use of gusset plates at joints was also adopted. All welding was carried out with Argonaut shielded-arc equipment.

The flooring consists of Noral corrugated aluminium sheeting of the deep-ridged "Industrial" pattern, 14 s.w.g., with the corrugations placed transversely. The sheets are carried on two longitudinal 3 in. by

2 in. by 3/16 in. plain angle sections welded to the lattice members, and are secured with aluminium rivets. The footway itself was made by laying about a ton of bituminous road material over the corrugated sheeting.

To reduce the spaces in the girders (to prevent a child, for example, falling through the side of the bridge) five 1 in. diameter aluminium tubes, the full length of the bridge, pass through the lattice members. Each tube is secured by welding only at each end of the bridge (where the handrails turn over to form the end posts), passing through clearance holes in the intermediate members.

The bridge, which was designed for Northern Aluminium Company, Limited, by Aluminium Laboratories, Geneva, was fabricated by the Atlantic Shipbuilding Company, Limited, at Newport. The complete structure (approximately 2 100 lb. without the bituminous decking) was transported to site by road, and launched with the help of two mobile cranes. At its points of support the bridge is carried on pin-joints, a link being provided at one end to allow for thermal expansion. When complete, the whole aluminium structure was left in its natural state, without painting or other protection. The fact that no painting will probably be required during its lifetime was, of course, one of the major factors leading to the choice of aluminium for its construction.

Tunnel ventilation-shaft inspection.

Hexagonal suspended cage and portable radio telephones.

(*The Railway Gazette*, March 15, 1957.)

To facilitate periodical examinations of tunnel vertical shafts by District Engineers, the equipment described below is in use on the Western Region of British Railways.

cat-head spanning the shaft at such a level as to permit of easy egress on to the top of the tower from a cage suspended from the cat-head. On the ground beside this stag-



Cage ascending; note steadying rope passing round anchor pulley from small winch.

Light military-type trestling is erected on each side of the tower forming the top of the shaft to be inspected. This supports a

ing is a 3 ton hand winch equipped with a 1/2-in. dia. non-rotating steel rope with a 9.9-ton breaking load, which passes through

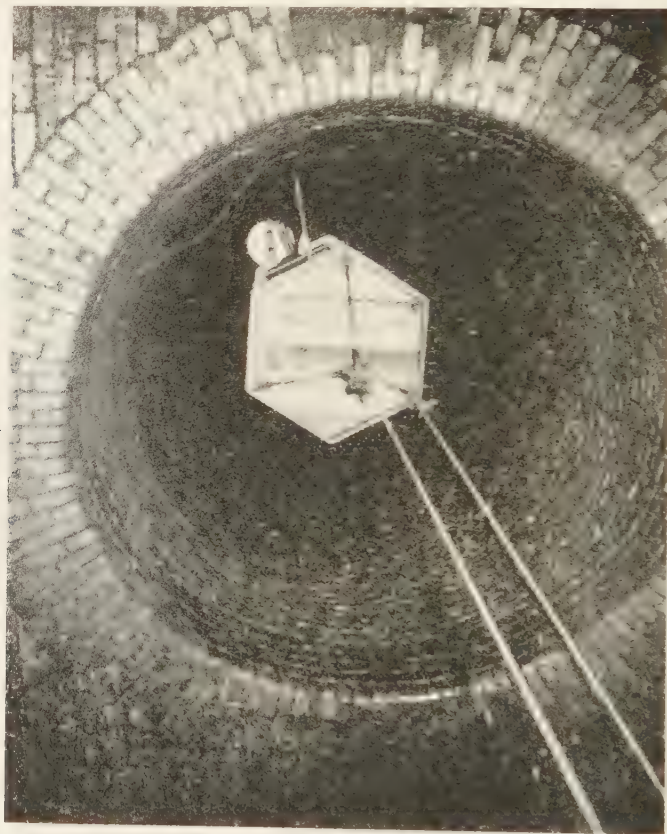
a pulley suspended from the centre of the cat-head. A swivel hook at the end of this rope carries the inspection cage.

Hexagonal cage.

This cage is hexagonal in plan, measuring 5 ft. 2 in. across the corners, and has a steel frame, a 2-in. thick timber floor, a

To record the height of the cage at any time above the crown of the tunnel, a hemp rope marked off at 10-ft. intervals hangs down from one corner of the underside of the cage. Communication between the top of the shaft, the cage, and the tunnel is maintained by three portable radio telephones.

The illustrations show this equipment in



Underside of cage with central steadying rope and corner measuring line.

roof, and a weight of 13 cwt. empty. It can accommodate three persons, but if walkie-talkie equipment is used it will carry only two in comfort. To the bottom of the cage a second rope is attached centrally, so that slight tension exerted upon it by a small winch in the tunnel steadies the cage and prevents excessive swaying.

use during the recent inspection of Cacrphilly Tunnel, 1933 yd. long, which has five 9-ft. dia. vertical aid shafts, measuring 142, 318, 362, 189, and 86 ft. from crown of tunnel to tops of towers.

We are indebted to Mr. M. G. R. Smith, Chief Civil Engineer, Western Region, for the above information.

The «Wire»... for raising track without a spotboard.

For years railroads have been lining curves with a string. Now they can raise track with a « wire ».

That's an informal name for a device that actually utilizes a wire, stretched on a moving assembly, to « sight » the track for a Tamping Power Jack.

(*Railway Track and Structures*, March, 1957.)

How it works.

Wire is stretched between the end carriages and through pointer mechanisms on the Tamping Power Jack. Scale at wire support on front carriage is set to indicate desired raise. A pointer on scale in front of operator on jack indicates amount of the raise at jacking location. Operator raises the grade rail until the pointer is slightly above the zero indication and brings the other rail to the correct cross elevation. The tie under the jack is then tamped to support the track.

An entirely new approach to the correction of track profiles is represented in a device being offered by the Nordberg Manufacturing Company. It has been named the « Trak-Surfacar ». In effect, the device takes the place of the conventional spotboard in track-raising operations. According to the manufacturer, it also eliminates, for all practical purposes, the necessity for running profiles and setting grade stakes.

What is it? How does it work? What are the results? These and other questions are answered below.

What is the principle on which this method is based?

This new track surfacing device was developed from the original ideas of W. C. McCormick, a general roadmaster on the Seaboard Air Line. The basic principle is to form a moving reference line with the top of the grade rail through the use of a steel wire, 125 ft. long, which is drawn tight.

What equipment is used when raising track by this method?

The central unit is a Nordberg Tamping Power Jack which raises the track, tamps

the tie at each jacking point to hold the raise and provides the propulsion power for itself and the reference wire. The wire is stretched between two four-wheel carriages, which are held apart from each other, and from the jack, by rods extending between a series of two-wheel buggies. Thus, the entire assembly is so constructed that it moves along the track as a unit.

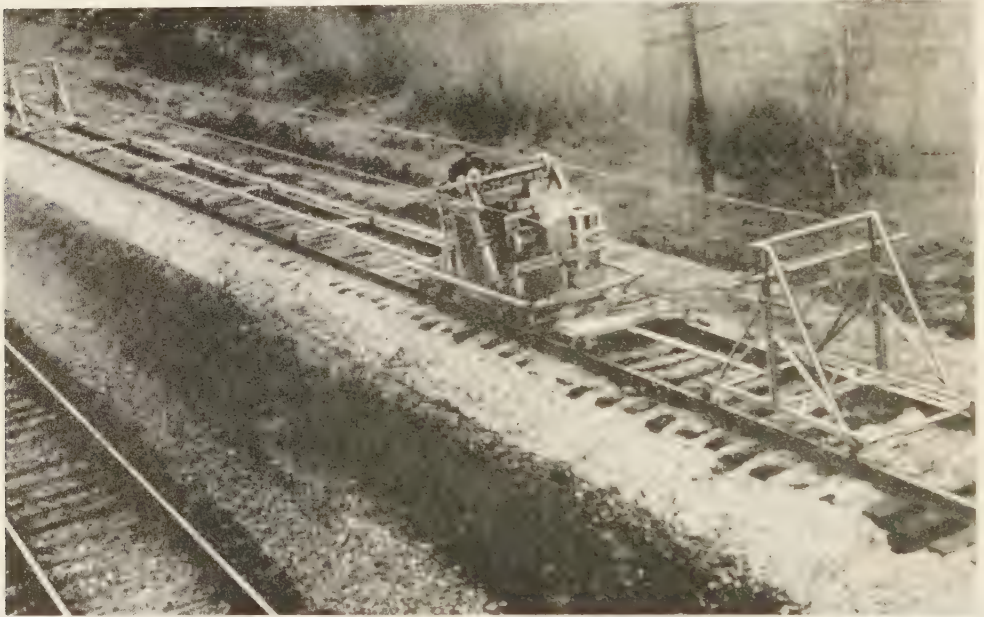
All the sections used in the assembly, including those in the front and rear carriages, are made of aluminum. It is reported that one man can remove any of the principal components from the track, and that two men can remove all sections in about two minutes.

The Tamping Power Jack is so located in the assembly that a cross-level mounted on it is placed 25 ft. from the rear carriage. The wire used is actually 250 ft. in length and extends between the carriages on both sides of the jack. Two pointers are mounted on the cross level on the jack directly in front of the operator. Either pointer can be used depending on which rail is used as the grade rail. Only the pointer in contact with the wire on this side of the jack is used. The other pointer is locked out of position. Thus, the grade rail is always raised to the wire with the opposite rail

being brought to correct cross level by means of the level bubble.

The manufacturer states that any Nordberg Tamping Power Jack can be used with the Trak-Surfacar without change. The Surfacar can also be used with the Nordberg Power Jack if desired. In that case a slight modification of the cross-level arrangement on the jack is made.

ing, the height of the wire above the top of the grade rail at the front carriage is exactly the same as the height of the wire above the grade rail at the rear carriage. This would be the setting if no raise were to be made. To make a raise, the wire adjustment on the front carriage is set for the lift desired and the rear adjustment is left at zero.



Entire assembly moves as unit. Direction of movement here is to the left.

Explain the procedure followed when using this equipment to raise track?

The equipment is assembled on the track as shown in the photograph reproduced above, with the direction of movement being toward the left. The wire, which passes through the pointer mechanism on the Tamping Power Jack, is held tight by a weight. The assembly is then moved to the first raising spot.

The wire supports on the front and rear carriages are set to scales by turning screws. When the adjustments indicate a zero read-

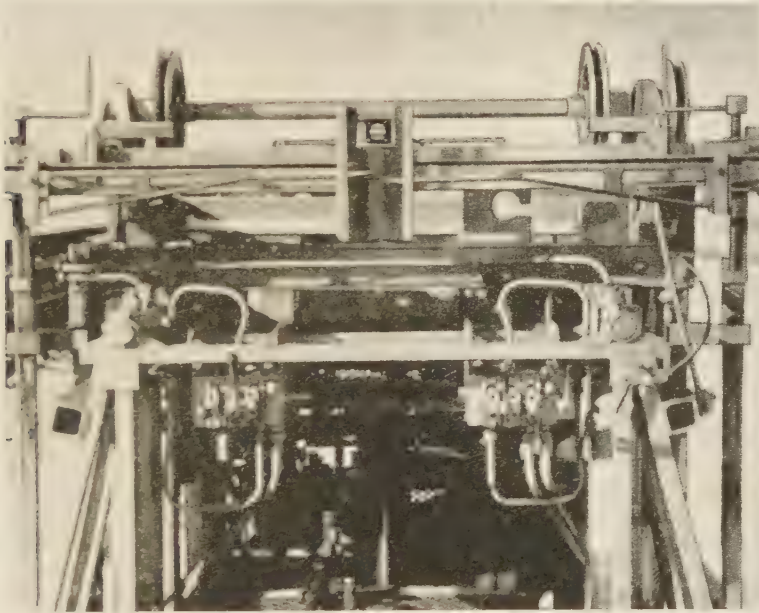
The jack operator observes the location of the pointer on the scale. If it is below the zero mark, he raises the grade rail until the pointer is slightly above the zero indication, and brings the opposite rail to correct cross elevation by observing the position of the level bubble on the jack cross level. The tie under the machine is then tamped to support the track in the raised position. The operator then releases the jack and observes the pointer again. If he has not allowed for enough settlement, he lifts the track again to make it right.

There is no "trick" in learning the use

of this equipment, according to the manufacturer. However, the operator of the Tamping Power Jack must take more than ordinary care to be certain that his grade rail pointer is as near zero as possible on the scale and that the cross level bubble is as perfectly centered as possible before he leaves one jacking location and moves to the next.

be titled like a track level. The same is true of the cross level mounted on the Tamping Power Jack. These tilting adjustments are used when raising track on a super-elevated curve. The low rail is ordinarily used as the grade rail.

When raising on curves, the wire support bars on the front and rear carriages must be kept level by lowering the side of the



Jack operator is governed by pointer and level bubble.

If the track is not settled enough to bring the pointer and the level bubble to correct position, the operator can move the machine slightly and bring the tamping bars down on the tie to « knock it down » until the pointer and level bubble give the correct indications. He then moves to the next jacking location and repeats the operation.

How is the equipment used when raising track on curves?

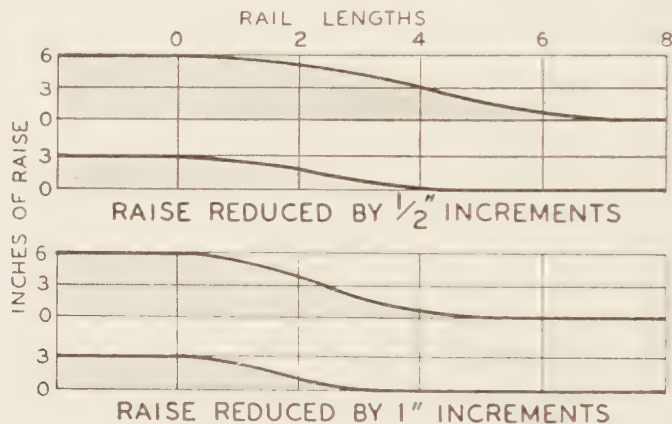
The wire support bars on the front and rear carriages are so mounted that they can

bar over the high rail. The super-elevation required at a particular jacking location is set by adjusting the cross level on the jack. This is done by lowering the end of the levelboard, which is over the high rail. Men are stationed at the front and rear carriages to accomplish the leveling of the wire support bars until full super-elevation is reached. When the spiral at the other end of the curve is reached by the front carriage, men must again be stationed at the carriages to keep the wire support bars level.

How are run-offs made?

When the last full raise is made, a man stationed at the front carriage lowers each side of the wire support bar in increments of 1 in. or 1/2 in. per jacking location. The length of the run-off desired determines the amount of the increment. The wire support bar on the front carriage is lowered until a reading of minus 2 in. is reached on the scales. It is said that this produces a safe and gradual run-off which can hardly be felt by a train passing over it.

of a sighting foreman. Thus, the mathematical relationship between the distances is 25 to 125, or 1 to 5. If the front carriage should be in a slight depression or should rest on a slight hump, this 1 to 5 ratio reduces the error at the jack to 1/5 of the irregularity on which the front carriage rests. Thus, while the method does not produce a perfectly engineered profile, for all practical purposes, it improves the existing profile five times, according to the manufacturer. The point is also made that



Curves show run-offs made in 1-in. or 1/2-in. increments.

What procedure is used when raising through switches?

Only a few frame or structural members of the assembly extend outside the rails. This is important when raising track through switches. Naturally, because of the interference of the turnout rails, neither the jack rams nor the tamping mechanism can be used. However, the wire and the pointer on the Tamping Power Jack can be employed in raising the switch, using hand jacks within the limits of the long timbers. It is reported that a No. 10 turnout was raised in 16 min. by this method.

Just what results are obtained with this method and equipment?

The Trak-Surfacar is, in effect, a constantly moving spotboard, with a distance of 100 ft. from the front carriage (or spot-board) to the jack, and 25 ft. from the jack to the rear carriage, which takes the place

the more frequently the track is raised using this method, the more closely the resulting profile approaches one which would be considered perfect by the engineer.

The manufacturer also points out that the Trak-Surfacar produces the desired effect on both sags and humps. In other words, as the wire passes over a sag, a greater lift is indicated than would be produced on level track. When the wire passes over a hump a smaller lift is indicated than would be the case if the track were level.

What this means is that the raise for which the device is set can be considered as being the "average" raise desired, since high spots within the length of the wire will be raised less than the setting and low spots will be raised higher than the setting. This is illustrated by experience on one railroad where a 1 3/4 in. raise setting produced some raises of only 1 1/8 in. and some as much as 3 1/2 in.

NEW BOOKS AND PUBLICATIONS.

[656 .28 (42)]

Lt.-Col. G.R.S. WILSON, C.B.E., Chief Inspecting Officer of Railways, Ministry of Transport and Civil Aviation. — **Report to the Minister of Transport and Civil Aviation upon the Accidents which occurred on the Railways of Great Britain during the year 1955.** — One brochure (6 × 9 1/2 in.) of 72 pages and graphs. — 1956, Her Majesty's Stationary Office, York House, Kingsway, London, W.C.2. and 423, Oxford Street, London, W.1. (Price : 3 s. net).

The framework of this report has remained unaltered. It groups the accidents into three classes : train accidents (derailments, collisions), traffic accidents (due to moving vehicles), and those which occurred on railway premises. In each case the victims are divided into : passengers, railway employees, other persons.

Detailed statistics are given in the report and condensed extracts inserted in the text.

Under the heading « General Summary », the author gives a table of figures relating to the years 1953, 1954, 1955 and the average figures for the years 1951 to 1955. The statistics are analysed separately for each of the three classes of accidents.

In the first class, the 18 cases which led to an enquiry are briefly reported, as well as others which were dealt with in agreement by the Regions. The author gives the measures decided upon to minimise human failures, or to improve the installations and stock. A special chapter is devoted to level crossings.

In the second class, as far as railwaymen are concerned, the author insists in particular on the need for protecting men working on the line. Class III groups those accidents which are not due specific-

ally to railway operating, but which have not had enough attention paid to them.

The various causes of accidents are gone into and examined in the chapter « Review of the year ». In the antiaccident campaign important factors are : the good state of repair of the track, the proposed use of lifting-up gates worked by track circuit, meticulous inspection of the rolling stock, and the investigation, which has already reached an advanced stage, into the use of an automatic train control system, and the rolling stock modernisation programme.

In his « conclusions » the author sums up his impressions of the year 1955. This was marred by several serious accidents, although fewer of these were due to the human element, and from other points of the statistics stand up well to a comparison with those for other years. In spite of these unfortunate accidents, the railway offers a very high degree of safety. Its employees are noteworthy for their professional skill and their sense of responsibility. No stone is left unturned to improve upon this high moral standard and to perfect the rolling stock and equipment.

E. M.

[621 .8]

PATIN (Pierre), Chief Engineer of the French National Railways. — **Les transmissions de puissance et la variation de la vitesse** (*The transmission of power and variation of the speed*). With a preface by M. A. CAQUOT, Member of the Institute. — One volume (6 1/4 × 9 7/8 in.) of 340 pages, with 318 figures and 8 tables. — 1956, Paris (V), Editions Eyrolles, 61, Boulevard St. Germain. (Price : 3 300 French francs).

M. PATIN's work is devoted to a study of the transmission of power, that is to say the whole of the components interposed between the generating component and the component using the power.

In the first part of his book, dealing with questions of a general nature, the author studies first of all, both as regards the traction transmissions and machine tool drives, the theoretical problem set to this transmission and establishes its ideal working conditions.

He then defines the various types of transmission used, then examines and gives a reasoned description of the auxiliary parts common to all transmissions, such as couplings, clutches, gears, reversing gears, plain or roller bearings, etc.

He ends this first part by a chapter dealing with the operation of the transmission, in which, after describing the characteristics of the servo-mechanisms, he describes mechanical, pneumatic, hydraulic and electric drives.

The second part of the book is devoted to the actual transmissions.

The author gives a painstaking description of the main types actually in use, dealing in turn with mechanical friction drives, mechanical gear drives, hydraulic drives, volumetric drives, hydromechanical drives, and D.C. and A.C. electric drives.

In each case, the author accompanies his description by the theoretical factors needed for designing and calculating the transmission in question.

To sum up, M. PATIN's book, which is easy to read and copiously illustrated with drawings, diagrams and photographs, will be a useful reference work in the libraries both of the user, to whom it will give all the information he requires about the working of the transmissions which he uses, and of the builder, to whom it will be a valuable aid in getting out his designs.

R. S.



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